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U. S. ARMORED CRUISER *TENNESSEE.*

BY LEWIS HOBART KENNEY, B. S., M. E., ASSOCIATE.

The contract for the construction of the *Tennessee* was signed with the William Cramp and Sons Ship and Engine Building Company, Philadelphia, Pennsylvania, on February 9, 1903. The time specified for the completion of construction of the ship was forty-two months and the contract price $4,035,000, of which $1,300,000 was apportioned for the machinery. The several events occurred during her construction as follows:

Keel laid,.....June 20, 1903;
Launched,.....December 3, 1904;
Dock trial of engines,.....August 24, 1905;
Official trials, left for.....February 1, 1906;
returned from, February 14, 1906.

There was no preliminary trial other than that made when under way for Boston, where the ship was to be dry docked and receive her supply of coal for the trial trips. The speed required by contract was 22 knots, and a penalty imposed of $50,000 if the speed should be below this by one-quarter knot.

PRINCIPAL DIMENSIONS OF HULL.

Length between perpendiculars, feet	502.0
on L.W.L., feet	503.5
over all, feet	504.5
Beam, extreme, feet	72.9
at L.W.L., feet	72.9
Ratio of length on L.W.L. to beam	6.91
Draught, designed, feet	25.0
Displacement, designed, tons	14,500
per inch immersion at L.W.L., tons	59.6
Area of immersed midship section, square feet	1,730
L.W.L. plane, square feet	25,040
Wetted surface, square feet	43,450
Coefficient of fineness, block	0.555
of M.S.	0.948
of L.W.L.	0.685
Cylindrical	0.585
Number of frames	123

ORDNANCE.

The following battery is provided:

MAIN BATTERY.

Four 10-inch B. L. rifles in two balanced electrically-operated turrets.

Sixteen 6-inch B. L. rifles, twelve on gun deck, and four on main deck.

SECONDARY BATTERY.

Twenty-two 3-inch (14 pounder) R. F. guns, twelve on gun deck, and ten on main deck.

Twelve 3-pounder S. A. guns, two on forward and four on after bridge and six on boat deck.

Two 1-pounder S. A. guns, aft on boat deck.

Two 3-inch field pieces on carriages.

Two machine guns of 0.30 caliber.

Four automatic guns of 0.30 caliber, two forward on boat deck and two for boats.

Two 18-inch submerged torpedo tubes located forward of the firerooms.

ARMOR.

Water-line belt, amidships, thickness, inches	5
ends, thickness, inches	3
Upper side, thickness, inches	5 and 2
Barbette, thickness, inches	7 and 4
Conning tower, thickness, inches	9
Protective deck, slope, pound plate	140
flat, pound plate	40
Splinter bulkheads on gun deck, pound plate	80

DESCRIPTION OF MACHINERY.

The following description of the machinery involves some of the data obtained from the trials conducted by the Board of Inspection and Survey and of tests made on auxiliary machinery. As this experimental data is subject to some error, due to the instruments used and the personal equation of the observers, the system used in computing from this data is:

To use:

3 significant figures when the error involved is equal to or greater than ..... 10.0 per cent.

4 significant figures when the error involved is ..... 1.0 per cent.
but less than ..... 10.0 per cent.

5 significant figures when the error involved is ..... 0.1 per cent.
but less than ..... 1.0 per cent.

6 significant figures when the error involved is ..... 0.01 per cent.
but less than ..... 0.1 per cent.

Hence it is unnecessary to record the experimental data with six significant figures when the error of the observed data is 1.0 per cent. Most of the data has been tabulated, and the curves may be interpolated to three significant figures.

The probable error may be less than 10.0 per cent., but the data is perhaps dependable to this degree.

MAIN ENGINES.

The main engines are vertical, inverted-cylinder, direct-acting, four-cylinder, triple-expansion, outboard-turning Navy Department design. There are two each in a water-tight compartment. The arrangement of cylinders is F.L.P., H.P., M.P. and A.L.P. Flanges are cast on the F.L.P. and H.P cylinders for bolting them together, and the same arrangement is used for the other two cylinders. The H.P. and M.P. cylinders have flanges which engage to prevent athwartship motion only. Fore-and-aft tie rods connect the two pair of cylinders. Athwartship tie rods through the center-line bulk head also connect the cylinders.

The frames are steel columns with X athwartship braces and in addition there are fore-and-aft and diagonal braces. The crank shaft is in two sections, with the F.L.P. and H.P. cranks 180 degrees apart, the M.P. and A.L.P. 180 degrees apart and the two sections 90 degrees apart. Crank sequence is thus: H.P., A.L.P., F.L.P. and M.P.

The turning engine is bolted to the outboard side of the main-engine bed plate, and operates a worm and wheel device. The reversing engine is on the inboard side, with trunnions on the steam cylinder which fit into bearings on the bed plate. It has one continuous piston rod through the steam- and oil-control cylinders to the reverse shaft. Stephenson double-bar links operate the piston valves. The points of cut off for these valves are graduated on the reverse-shaft arms, and are the mean of the top and bottom.

Jacket steam passes through the top, sides and bottom of each cylinder. It is supplied from the main steam line through a reducing and relief valve to the H.P. cylinder, and through a reducing and relief valve from each cylinder to the succeeding one. Jacket drains from the L.P. cylinders are to the main feed tanks through expansion traps.

The handling gear for the engines is on the inboard side, and directly back of it on the center-line bulkhead are the various gauges and indicators.

BED PLATES.

The bed plates are in three sections. They are steel castings as also are the main bearing caps. The bottom brasses are circular to facilitate removal. Parsons white brass is used to line the bearings which are cast hollow for water service. A cross section of bed plate is generally of the I form.

DATA FOR BED PLATE.

Number of sections per engine	3
bearings per engine	6
Main bearing bolts, number per bearing	4 and 2
diameter, inches	$4\frac{7}{16}$
Vertical web, width, inches	$1\frac{3}{4}$
height, inches	16
Coupling bolts, diameter, inches	$1\frac{1}{4}$
number per coupling	12
Length of bearings, inches	$29\frac{1}{2}$ and 17

COLUMNS.

The cylinders are supported by forged-steel columns which are strengthened, athwartship, by X braces and diagonal fore-and-aft tie rods. A strong back, I section, of cast steel extends longitudinally on the inboard side of each engine to support the lower ends of the crosshead guides.

DATA FOR COLUMNS.

Number per engine	12
Diameter, inches	$6\frac{1}{2}$
Bolts, number per top and bottom flange	4
diameter, inches, top flange	$2\frac{1}{4}$
bottom flange	$2\frac{1}{4}$

CYLINDERS.

The cylinders are cast iron fitted with hard close-grained cast-iron liners for the pistons and in valve chests. In each cylinder cover is a manhole and an additional one

in the bottom of the M.P. and L.P. cylinders. All cylinders are clothed with a carbonate of magnesia and asbestos fiber compound and lagged with Russian sheet iron.

The clearance volume for the cylinders was determined in the shop. For this purpose the cylinders were placed horizontal with the valve chest uppermost. The piston was placed at its extreme position and braced firmly in place. Previous to this the packing rings of the piston had been removed and soft packing substituted and the piston valves entirely removed, in their place was fitted cast-iron packing rings with glands at each end, and made, with soft packing, a perfectly tight joint.

This position of the cylinder placed the valve sight holes at the top, and they were used to admit the water and for the free exit of air. By this method there were no air pockets to introduce an error into the observations. On the gallery of the machine shop was a tank, on a set of scales, filled with water by a pipe from the main floor. A branch from this pipe led to the cylinder sight hole. This pipe acted as a syphon to fill the cylinder from the tank and gave the correct weight of water required to fill the clearance space without corrections for weight of pipe, etc. The temperature of the water was noted and the clearance volume computed from this data. After the engine had been installed the piston linear clearance was determined and the corresponding volume computed. This volume, added to the volume determined as above, gave the true clearance volume.

The index plates for this data were placed on the inboard side of the crosshead guides.

DATA FOR CYLINDERS.

	H.P.	M.P.	L.P.
Cylinders, number	1	1	2
diameter, inches	38½	63½	74⅛
ratio of areas	1	2.72	7.42
Stroke, inches	48	48	48
Thickness of body inches	2¼	2	1⅞
liner, inches	1¾	1⅝	1¼
jacket space, inch	⅝	⅝	⅝
valve chests, inches	1⅝	1½	1⅜
Cylinder cover studs, number	48	60	42
diameter, inches	1⅝	1⅜	1⅛
pitch, inches	3.6	4.8	7.1
Valve-chest cover studs, number	28	26	18
diameter, inches	1¼	1⅛	1⅛
pitch circle, diameter, ins.	33¼	32½	34½
Piston-liner tap-bolts, number	32	36	42
diameter, inch	⅞	⅞	⅞
pitch circle, diameter, ins.	36½	61½	72
Nozzles, steam, diameter, inches	15	(2)—14	(2)—15
exhaust, diameter, inches	(2)—14	19½	28
relief valve, diameter, inches	4½	4½	4½
jacket steam, diameter, inches	1¼	1¼	1

	Piston linear clearance, inches.	Piston displacement, cu. in.	Clearance volume, cu. in.	Per cent. clearance.
Port engine.				
H.P., top	1 1/16	55,870	17,840	32
bottom	1 3/16	53,130	18,490	35
M.P., top	7/8	152,000	34,850	23
bottom	1 1/16	149,300	37,190	25
F.L.P., top	7/8	207,100	45,410	22
bottom	7/8	204,300	41,230	20
A.L.P., top	7/8	207,100	45,410	22
bottom	7/8	204,300	41,230	20
Starboard engine.				
H.P., top	1 1/16	55,870	17,840	32
bottom	7/8	53,130	18,420	35
M.P., top	1 7/16	152,000	34,260	23
bottom	7/8	149,200	36,990	25
F.L.P., top	7/8	207,100	45,950	22
bottom	1 9/16	204,300	40,440	20
A.L.P., top	1 1/16	207,100	45,140	22
bottom	1 1/16	204,300	40,970	20

MAIN ENGINE VALVES.

Piston valves, single ported, are fitted in each cylinder. The body of the H.P. valve is cast iron and of the others steel pipe, flanged, to which are bolted cast-iron heads. The packing rings are hard, close-grained cast-iron slotted and fitted, inside, with a clamp to prevent expansion be-

yond a prescribed limit. Forged-steel follower rings are used on all valves. There is a balance piston in the valve-chest cover for each valve. The H.P. valve stem is connected direct to the link-block pin but the M.P. and L.P. are to a cast-steel crosshead which in turn is connected to the link-block pin. The reverse-shaft levers have been graduated for five points of cut off, which were determined by observations made after the engines had been installed, and are the mean of top and bottom determinations.

DATA OF INDEX OF LEVERS.

Port engine.				
H.P. cylinder	.78	.72	.67	.60 and .53
M.P. cylinder	.76	.71	.65	.58 and .51
F.L.P. cylinder	.73	.67	.62	.56 and .50
A.L.P. cylinder	.73	.68	.63	.56 and .48
Starboard engine.				
H.P. cylinder	.78	.73	.68	.61 and .53
M.P. cylinder	.76	.71	.65	.58 and .50
F.L.P. cylinder	.73	.68	.62	.57 and .51
A.L.P. cylinder	.73	.68	.62	.56 and .49

DATA FOR VALVES.	H.P.	M.P.	L.P.
Packing ring, width, axially, top, inches	$5\frac{7}{16}$	$5\frac{3}{32}$	$6\frac{1}{32}$
bottom, inches	5	$5\frac{1}{32}$	7
radially, inch	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
Balance piston, diameter, inches	6	6	10
Valve stem, diameter through gland, inches	$3\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{2}$
valve, inches	2	2	2
crosshead area, square inches, each side	27	67	67
Link-block pin, diameter, inches	5	5	5
length, inches	$4\frac{1}{2}$	$4\frac{1}{2}$	$4\frac{1}{2}$
Eccentric, diameter, inches	34	34	34
width of face, inches	$5\frac{1}{2}$	$5\frac{1}{2}$	$5\frac{1}{2}$

Measurements were made of the valve settings after the engines had been installed and had had a dock trial. Both engines were measured for four positions: full, mean and minimum gear ahead and full gear backing. Valve diagrams were drawn from this data. Below is the data for three positions of the valves of the port engine:

PORT ENGINE.—VALVE DIAGRAM DATA.

Name of vessel..... U. S. A. C. *Tennessee.*

Total I.h.p. and displacement.....23,000 I.h.p. 13,500 tons.

Type and number of engines.....Two 4-cylinder, inverted, triple expansion.

Diameters of cylinders and stroke.....38½ by 63½ by 74½ by 74½ by 48 inches.

Valve gear.....Stephenson double-bar links, open rods.

Revolutions per minute designed.....130.

Piston speed per minute.....960.

Connecting rod.....length between centers = 96 inches; ratio to crank = 4.

	H.P.		I.P		F.L.P.		A.L.P.	
Eccentricity, inches	5.25		5.25		5.25		5.25	
Travel of valve, inches	10.50		10.50		10.50		10.50	
Type, number, and diameter if piston valves	1—26		2—26		2—28		2—28	
Side of valve on which steam is taken	Inside.		Outside.		Outside.		Outside.	
	Top	*Bottom*	*Top*	*Bottom*	*Top*	*Bottom*	*Top*	*Bottom*
Width of port, inches	3.38	3.38	3.69	3.69	4.00	4 00	4.0	4.00
Steam lap, inches	2.38	1.69	2.31	1.81	2.06	2.06	2.13	2.13
Exhaust lap, inches	—0.50	—0.09	—0.31	—0.09	+0.25	+0 94	+0.31	+0.69
Angular advance, degrees	34.25		36.00		39.25		39.38	
Steam lead, angular, degrees	7.8	14.5	9.0	15.3	16.0	16.0	15.3	16.0
linear, inches	0.63	1.19	0.69	1.19	1.25	1.25	1.19	1.25
Cut-off in inches. (A, see fig.)	37.6	36.9	37.5	35 0	37.5	32.6	37.4	32.2
decimal of stroke. (A, see fig.)	0.783	0.769	0.782	0.729	0.782	0.679	0.780	0.672
Release in inches. (B, see fig.)	44.0	42.9	43.5	41.8	44.3	43.6	44.3	43.4
decimal of stroke. (B, see fig.)	0.917	0.895	0.905	0.874	0.925	0.908	0.924	0.905
Compression in inches. (C, see fig.)	3.3	2.9	4.9	3.5	7.6	6.3	8.4	6.1
decimal of stroke. (C, see fig.)	0.069	0.060	0.102	0.073	0.158	0.131	0.175	0.127
Steam opening, linear, from diagram, inches	2.9	3.4	2.8	3.1	3.2	3.2	3.1	3.1
area of, in square inches	178	209	348	380	436	436	428	428
Exhaust opening, linear, from diagram, inches	3.4	3 4	3.7	3.7	4.0	4.0	4.0	4.0
area of, in square inches	209	209	457	457	545	545	545	545
Steam velocity through ports in feet per minute	6,280	5,080	8,740	7,860	9,520	9,370	9,650	9,550
Exhaust velocity through ports in feet per minute	5,350	5,080	6,660	6,540	7,600	7,510	7,600	7,520

Velocity through main steam pipe in feet per minute.....6,340.

Velocity of exhaust to condenser in feet per minute.....6,730.

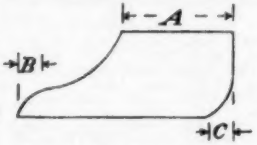
(Full gear ahead.)

	H.P.		I.P.		F.L P.		A.L.P.	
Eccentricity, inches	5.25		5.25		5.25		5.25	
Travel of valve, inches	8.00		8.13		8.44		8.44	
Type, number, and diameter if piston valves	1—26		2—26		2—28		2—28	
Side of valve on which steam is taken	Inside.		Outside.		Outside.		Outside.	
	Top	*Bottom*	*Top*	*Bottom*	*Top*	*Bottom*	*Top*	*Bottom*
Width of port, inches	3.38	3.38	3.69	3.69	4.00	4.00	4.00	4.00
Steam lap, inches	2.31	1.63	2.95	1.81	2.03	2.16	2.13	2.06
Exhaust lap, inches	—0.56	—0.01	—0 22	—0.17	+0.38	+0.69	+0.28	+0.78
Angular advance, degrees	57.50		58.88		62.88		62.50	
Steam lead, angular, degrees	22.0	33.5	25.8	31.5	33.0	33.0	32.0	33.3
linear, inches	1 06	1.75	1.25	1.63	1.69	1.63	1.63	1.69
Cut-off in inches. (A, see fig.)	25.9	24 8	25.8	23.0	26.6	19.1	26.0	20.3
decimal of stroke. (A, see fig.)	0.540	0.517	0.537	0.479	0.554	0.398	0.542	0 423
Release in inches. (B, see fig.)	36.5	34.1	37.8	33.6	39 0	36.4	38.8	36 9
decimal of stroke. (B, see fig.)	0 761	0.710	0.787	0.700	0.813	0.759	0.808	0.769
Compression in inches. (C, see fig.)	10.3	9.3	12.5	8.5	17.5	13.9	16.9	14.3
decimal of stroke. (C, see fig.)	0.214	0.194	0.260	0.177	0.365	0.290	0.352	0.298
Steam opening, linear, from diagram, inches	1.7	2.4	1.8	2.3	2.2	2.1	2 1	2 2
area of, in square inches	103	148	223	285	300	286	286	300
Exhaust opening, linear, from diagram, inches	3 4	3.4	3.7	3.7	4.0	4.0	4.0	4.0
area of, in square inches	209	209	457	457	545	545	545	545

(Minimum gear ahead.)

	H.P.		I.P.		F.L.P.		A.L.P.	
Eccentricity, inches	5.25		5.25		5.25		5.25	
Travel of valve, inches	10.50		10.50		10 50		10 50	
Type, number and diameter if piston valves	1—26		2—26		2—28		2—28	
Side of valve on which steam is taken	Inside.		Outside.		Outside.		Outside.	
	Top	*Bottom*	*Top*	*Bottom*	*Top*	*Bottom*	*Top*	*Bottom*
Width of port, inches	3.38	3.38	3.69	3.69	4.00	4.00	4.00	4.00
Steam lap, inches	2.31	1.63	2.19	1 94	2 00	2.25	2.09	2.13
Exhaust lap, inches	—0.44	+0.06	—0.03	—0.38	+0.44	+0.69	+0.22	+0.75
Angular advance, degrees	34.50		35.75		35.94		35.50	
Steam lead, angular, degrees	8.8	16.0	11.0	14.0	12.3	11.3	11.8	11.8
linear, inches	0.69	1.31	0.88	1.13	1.00	0.88	0.94	0.94
Cut-off in inches. (A, see fig.)	38.0	36 8	38.1	34 6	39.0	32.6	38.6	33.9
decimal of stroke. (A, see fig.)	0.792	0.766	0.794	0.721	0.813	0.679	0.805	0.707
Release in inches. (B, see fig.)	42.3	42.9	44.1	40.6	45.3	44.8	45.1	44.6
decimal of stroke. (B, see fig.)	0.884	0.895	0.919	0.845	0.945	0.935	0.939	0.928
Compression in inches. (C, see fig.)	3.5	3.5	6.0	3.0	7.4	4.9	6.8	5.3
decimal of stroke. (C, see fig.)	0.073	0.073	0.125	0.063	0.154	0.102	0.142	0.110
Steam opening, linear, from diagram, inches	2.9	3.4	3.1	3.3	3.3	3.0	3.2	3.1
area of, in square inches	178	209	384	408	450	408	436	422
Exhaust opening, linear from diagram, inches	3.4	3.4	3.7	3.7	4.0	4.0	4.0	4.0
area of, in square inches	209	209	457	457	545	545	545	545

Positions of pistons to be given as follows: **(Full gear astern.)**



Scales, crank circle, 3 in = 1 ft.
valve circle, full size.

REMARKS.

To obtain the above data from the valve diagrams a separate diagram was drawn for each end of each cylinder. The "angular advance" obtained is the mean for the top and bottom ends of each cylinder.

DATA FROM OFFICIAL TRIALS.

	Port engine. Steam.	Exhaust.
Velocity in feet per minute at maximum revolutions per minute (130):		
H.P. valve, port, top	6,640	5,660
bottom	5,660	5,660
M.P. valve, port, top	9,380	7,140
bottom	8,590	7,140
L.P. valve, port, top	10,200	8,180
bottom	10,200	8,180
Main steam pipe	6,640	.....
exhaust pipe	.....	7,240
Velocity of valve in feet per minute	.....	227
Velocity in feet per minute at average revolutions per minute, for 4-hour trial (127.4):		
H.P. valve, port, top	6,510	5,540
bottom	5,540	5,540
M.P. valve, port, top	9,210	7,010
bottom	8,440	7,010
L.P. valve, port, top	10,000	8,020
bottom	10,000	8,020
Main steam pipe	6,510	.....
exhaust pipe	.....	7,100
Velocity of valve, in feet per minute	.....	223

Velocity in feet per minute, at 100 revolutions per minute:	*Port engine.* Steam.	Exhaust.
H.P. valve, port, top	5,110	4,350
bottom	4,350	4,350
M.P. valve, port, top	7,230	5,500
bottom	6,620	5,500
L.P. valve, port, top	7,850	6,300
bottom	7,850	6,300
Main steam pipe	5,110	.....
exhaust pipe	.....	5,570
Velocity of valve, in feet per minute	.....	175

PISTONS.

All pistons are conical-shaped steel castings fitted with cast-steel follower rings, which are held in place by countersunk collar studs. There are two hard cast-iron packing rings for each piston, which are slotted and clamped, inside, to prevent expansion beyond a prescribed limit, and are held against the cylinder wall by springs. The top of the piston is threaded for a nut to be used when removing the piston from its rod.

DATA FOR PISTONS.

Packing rings, number per piston	2
width, each, axially, inches	1¼
radially, inches	1
Follower studs, number, H.P. piston	12
M.P. piston	20
L.P. piston	20
diameter, inches	1¼

DATA FROM FOUR-HOUR TRIAL.

Mean piston speed, in feet per minute, at 127.4 R.p.m	1,019
Maximum piston speed, in feet per minute, at 130 R.p.m	1,140

PISTON RODS, CROSSHEADS AND CROSSHEAD SLIPPERS.

The piston rods are made of open-hearth nickel-steel annealed and oil tempered, but the crossheads are of a slightly lower grade of steel. The crossheads slippers are of Parsons manganese-bronze castings faced with white brass. Axial holes are bored through all piston rods and into the ends of the crosshead pins. The piston rods,

ground to the designed diameter, are tapered and threaded, for a nut at each end to fit the piston and crosshead. The crosshead guides are cast iron, hollow for water service, attached to a projection from the cylinder at the top and a cast-steel I beam between columns at the bottom. The backing guides are also cast iron.

DATA FOR PISTON RODS, CROSSHEADS AND CROSSHEAD SLIPPERS.

	H.P. and M.P.	L.P.
Piston rod, diameter, inches	8½	8½
axial hole, inches	4½	5
Crosshead pin, diameter, inches	11½	9½
length, inches, (each)	12⅛	10⅛
of brass, inches	12½	10½
Slipper, length, inches	34	34
width, inches	27	18
area, in square inches	918	612
width, backing (each), inches	9½	6½
area, backing, square inches	646	444
ratio of above areas	.70	.73

DATA FROM TRIALS.

Calculated from receiver pressures of port engine.

	H.P.	M.P.	L.P.
Mean pressure in pounds, per square inch of crosshead slipper area at:			
127.4 R.p.m., mean of four-hour trial	52	54	70
100.4 R.p.m., from standardization trial	36	28	35
Mean pressure in pounds, per square inch of crosshead pin projected area at:			
127.4 R.p.m.	660	690	860
100.4 R.p.m.	460	360	430
Per cent. cut off of valves at:			
127.4 R.p.m.	78	76	73
100.4 R.p.m.	78	76	73
Indicated horsepower at:			
127.4 R.p.m.	4,361	4,029	4,556
100.4 R.p.m.	2,350	1,910	1,990
Receiver pressure pounds, guage, at:			
127.4 R.p.m.	250	90	27
100.4 R.p.m.	155	40	7
Speed in knots, at:			
127.4 R.p.m.	...	...	22.16
100.4 R.p.m.	...	...	18.4

CONNECTING RODS AND CRANK SHAFTS.

The connecting rods are open-hearth nickel-steel, annealed and oil-tempered forgings; but the crank shafts are a slightly lower grade of steel. Axial holes are bored through rods and shafts. The upper ends of the rods are forked for the crosshead-pin brasses. All brasses are lined with Parsons white brass. The crank shafts are in two sections assembled at an angle of 90 degrees. The sequence of cranks—interval 90 degrees—is H.P., A.L.P., F.L.P. and M.P.

DATA FOR CONNECTING RODS AND CRANK SHAFTS.

	H.P. and M.P.	L.P.
Connecting rod, length between centers, inches	96	96
diameter, top, inches	7¼	6¼
bottom, inches	8¾	7¾
axial hole, inches	2	2
crank-pin bolts, diameter, inches	4 1/16	4 1/16
crosshead-pin bolts, diameter, inches	3 7/16	2 1/16
Crank shaft, diameter, inches	...	18¼
axial hole, inches	...	10
crank pin, diameter, inches	...	20
axial hole, inches	...	12
length, inches	...	23
brass, inches	...	22½
webs, thickness, inches	...	21½
width, inches	...	12
coupling flange, diameter, inches	...	32
width, inches	...	4½
bolts, number	...	12
diameter, inches	...	3¼

DATA FROM TRIALS.

Mean pressure, in pounds per square inch, of projected area of crank pin:		
R.p.m.	127.4	100.4
H.P.	368	258
M.P.	385	202
L.P.	333	167
Mean velocity, in feet per minute, of rubbing surface in bearing:		
Crank pin	667	524
Main bearing	616	485

SHAFTING AND THRUST BEARINGS.

All shafts are steel forgings the next grade below the open-hearth nickel-steel, oil tempered and annealed used for the piston and connecting rods. The shafts are in three sections: thrust, stern-tube and propeller.

The shaft slopes upward on its way aft .1585 inch per foot, and diverges from the center line at the rate of .1646 inch per foot. A clip coupling is used for the stern tube to the thrust shaft, which consists of a forged-steel ring, prevented from rotating by four keys. Forward of this ring are two half rings which fit into a circumferential groove in the stern-tube shaft. The coupling bolts pass through the forged-steel ring, the two half rings and the flange of the thrust shaft.

The outboard coupling for the stern tube and propeller shaft is a steel forging machined to receive the tapered ends of the two shafts, which are held in place by axial and diametral keys. This coupling is covered by a fair-water casing filled with a compound of tallow and rosin, which when cold forms a thick viscous mass.

The stern tube is encased in brass and the space between it and the shaft is filled with this tallow-rosin mixture.

The body of the thrust bearing is cast iron designed to form a reservoir for oil and a bearing at each end for the thrust shaft. These are fitted with a stuffing box to prevent the leakage of oil. The thrust shoes are cast steel of the usual horse-shoe type, hollow for water service and faced with Parsons white brass. The body sets on the sole plate, and at each end of this is an arrangement of wedges to provide for axial adjustment. A cover of steel plate completely protects the thrust bearing. This has hinged covers to facilitate examination of bearing while under way. At the bottom of the reservoir are brass pipes for water service.

DATA FOR SHAFTING.

Thrust shaft, diameter, inches	17½
at bearing, inches	18½
axial hole, inches	10½
collars, number	13
outside diameter, inches	28
width, inches	2
space, inches	4½
coupling bolts, number, forward flange	12
after flange	10
diameter, forward flange, inches	3¼
after flange, inches	3½
Stern-tube shaft, diameter, inches	18½
at bearings, inches	20¼
axial hole, inches	10½
length, forward bearing, inches	58½
after bearing, inches	58½
sleeve coupling, diameter, outside, inches	24
length of taper, inches	35
axial key, inches	2½ by 1½
diametral key, inches	8 and 7 by 2½
Propeller shaft, diameter, inches	18½
at bearings, inches	20¼
axial hole, inches	10½
length forward shaft bearing, inches	39½
after shaft bearing, inches	61½
threads, per inch, propeller nut	4
outside diameter, inches	12

DATA FOR THRUST BEARING.

Thrust shoes, number	12
total area, in square inches	2,810
Thrust-shaft bearings, length, inches	18½
Thrust side rods, diameter, inches	4

DATA FROM TRIALS.

R.p.m	127.4	100.4
Indicated thrust, in pounds, per sq. in. on thrust-bearing surface.	69	39
Mean velocity, in feet per minute, of rubbing surface	780	615

FORMULA.

$$T = \frac{\text{I.h.p.} \times 33{,}000}{V}$$

Symbols:

T. = indicated thrust in pounds.
I.h.p. = indicated horsepower of main engines.
V. = velocity of ship in feet per minute.

PROPELLERS.

The propeller blades and hubs are Parsons manganese-bronze.

The hub is fitted on the tapered end of the propeller shaft and held in place by one longitudinal key and a nut at the end of the shaft, threaded in the opposite direction to the rotation of the propeller. The starboard propeller rotates in a clock-wise direction for ahead motion. There are recesses in the hub into which the blades are fitted and held by rolled manganese-bronze tap bolts; a small casting is placed between bolts to prevent their turning. The bolt holes in the blades are elongated to allow for adjusting the pitch, which is variable, due to the working surface being slightly convex. The propellers were carefully balanced, finished all over and burnished.

The pitch of the propeller blades was carefully determined in the shops for the maximum, mean and minimum positions of the blade in the hub. Graduations for these three positions were placed on the base of each blade and a reference mark made in the hub. The per cent. slip is computed from the mean pitch and the speed of the screw, and the indicated thrust as for the thrust bearing.

DATA FOR PROPELLERS.

Number of blades	3
Diameter, feet	18
Pitch, mean, as set, feet	22
maximum, feet	23.5
minimum, feet	20.5
Ratio of diameter to pitch	1.22
Area, square feet, helicoidal	92
projected	74
disc	254
immersed midship section	1,727
ratio, projected to disc area	0.291
helicoidal to disc area	0.362
disc to immersed midship section	0.147
projected to immersed midship section	0.0429
helicoidal to immersed midship section	0.0534
Tap bolts, number per propeller	27

Tap bolts, diameter, inches	4
length under head, inches	7⅞
Longitudinal key, length, inches	40
width and depth, inches	2½ by 4½
Immersion of upper tip of blade, inches	69
Height of lower tip of blade above keel, inches	15
Test of propeller material, tensile strength, pounds	68,400
elastic limit, pounds	33,800

DATA FROM STANDARDIZATION TRIALS.

Revolutions per minute.	Indicated horsepower.	Knots.	Per cent. slip.	Indicated thrust, pounds, per sq. in.		
				Disc.	Helicoidal.	Projected.
90	8,600	16.74	14.3	2.3	6.4	7.9
100	12,000	18.36	15.2	2.9	8.0	10.0
110	16,200	19.94	16.5	3.7	10.0	12.5
120	21,600	21.28	18.4	4.5	12.5	15.5
*127.4	26,500	22.16	19.9	5.3	14.7	18.3

* From four-hour trial.

TURNING ENGINE.

The turning engine is bolted to the outboard side of the main engine bed plate just abaft the coupling of the crank shaft. Around this coupling is the large worm wheel which engages a worm on an inclined shaft. The worm is held in place by a feather and a diametral key above, but by removing the key the worm may be moved axially, to disengage it, on the shaft and the key inserted below to keep it in this position. At the lower end of this shaft is a thrust bearing of three composition washers, and at the upper end is a worm wheel which engages the worm on the crank shaft of the turning engine. This worm shaft has a roller thrust bearing. A plug cock is designed to reverse the engine by reversing the direction of the steam and exhaust.

The end of this crank shaft is made square to fit a hand-operated wrench, and a small Gypsy head is fitted to it in order that the turning engine may be used as a winch for handling heavy weights in the engine room.

DATA FOR TURNING ENGINE.

Cylinders, number	2
diameter, inches	6
stroke, inches	7
Piston rod, diameter, inches	1⅝
Valve, piston, diameter, inches	3
travel	1½
Crank shaft, diameter, inches	2½
number of bearings	3
length of bearings, inches	4 and 6
crank pin, diameter, inches	2½
length, inches	3½
crank angle, degrees	90
Crosshead slipper, area, square inches	17
pin, diameter, inches	1⅝
length, inches	3
Connecting rod, length between centers, inches	14
diameters, inches	1⅝ and 1½
Worm, small, diameter, pitch circle, inches	4½
wheel, diameter, pitch circle, inches	14½
teeth, number	30
Worm, large, diameter, pitch circle, inches	9.7
wheel, diameter, pitch circle, inches	57.3
teeth, number	60
shaft, diameter, inches	4
Step bearing, area, square inches	26
Pipes, diameter, steam, inches	1½
exhaust, inches	1½

REVERSING ENGINE.

The reversing engine is supported by two trunnion bearings on the inboard side of the main-engine bed plate. The trunnions on the steam cylinder are to allow for oscillatory movement of the engine. Above this cylinder is the oil-control cylinder, and extending through both is the piston rod, which is connected direct to the reverse shaft crank levers. Valves on the oil-control cylinders automatically by-pass the oil when using steam, or they may be used to connect to the hand oil pump.

Near the main engine throttle-valve hand wheel is a hand lever to operate the differential valve mechanism of the engine.

The reverse shaft is an open-hearth, nickel-steel, oil-tempered and annealed forging with a hole bored axially. This is on the inboard side of the engine near the top of the columns.

DATA FOR REVERSING ENGINE.

Cylinder, steam, diameter, inches	16
trunnions, diameter, inches	7
length, inches	7¼
oil, diameter, inches	10
stroke, common, inches	22
Piston rod, diameter, inches	4
crank pin, diameter, inches	5
length, inches	5
levers, length, inches	17
Hand oil pump, piston, diameter, inches	2
stroke, inches	5
Reverse shaft, diameter, inches	8½ and 5½
axial hole, inches	5 and 2½
bearings, number	6
diameters, inches	8½ and 5½
length, inches	9
Pipe, steam, diameter, inch	¾
exhaust, diameter, inch	1

OIL AND WATER SERVICES.

At the after end of the engine rooms are the oil reservoir tanks, from which oil is drawn by oil pumps, on the center-line bulkhead above the upper gratings, and discharged to distributing tanks. These pumps can draw from or discharge to any port or starboard tank. The oil flows from the distributing tanks, by gravity, to the oil boxes and oil cans of the main and auxiliary machinery in the engine rooms. Sight and wick feeds are fitted in all oil boxes. The crosshead and crank pins are bored for oil reservoirs to which oil is transmitted by pipes and banjo oilers, respectively. Extensions from the water service introduce water to these reservoirs which causes the oil to saponify. At the lower end of the crosshead guides is an oil reservoir into which a scoop, on the lower end of the crosshead slipper, dips and distributes the oil on its up stroke. The eccentrics are oiled by a system of telescopic tubes, and the thrust bearings by filling the body with oil and water. In addition to this there are oil cups at the top of each thrust shoe fitted with wick feed.

The water-service supply is from a nozzle on the main circulating-pump discharge to the main condenser. This pipe is led below the floor plates to the inboard side of the main engines, with extensions to manifolds on the engine columns.

The valves of these manifolds distribute water to the following places:

Main bearing, through cap and brass.
on cap.
Crosshead, slipper.
guides.
pin, to oil reservoir.
on brasses.
Crank pin, to oil reservoir.
on brasses.

There is also a water service to the eccentrics, thrust bearings and stern tubes.

CRANK PIT DRAINAGE.

The crank pits are arranged to drain forward, through limber holes in the foundations, to the suction well of the shaft bilge pump. This pump is driven from a crank on the forward end of the main-engine crank shaft. The pump discharges directly overboard and has an additional suction to the engine-room bilge drainage. The fire and bilge pump has a valve, in its suction manifold, to the crank-pit suction well.

DATA FOR SHAFT BILGE PUMP.

Diameter of plunger, inches	10
stroke, inches	6

BOILERS.

The boilers installed are the usual Babcock & Wilcox type which have been described in detail in the JOURNAL by Lieutenant R. K. Crank, U. S. N., in volume XVI, page 1223, and Commander W. N. Little, U. S. N., in volume XVII, page 191. The firerooms are divided into eight

watertight compartments with two boilers per compartment. Shaking grates of the contractor's design are installed.

DATA FOR BOILERS.

Number, per ship	16
Total heating surface, square feet	70,940
grate surface, square feet	1,650
Ratio, H.S. to G.S.	43
Smoke pipes, number	4
height above grates, feet	92
area of cross section, square feet	60
number of boilers to each	4
Ratio of G.S. to smoke-pipe area	6.9
Headers, number per boiler	27
Tubes, number of 2-inch, per middle headers	34
4-inch, per middle headers	2
length between headers, feet	8
Fireroom, length over all, feet	136
width, feet	23
total floor space, square feet	6,280
Total boiler floor space, square feet	2,340
Ratio of above	0.372
Volume of firerooms, cubic feet	135,600
boilers, cubic feet	32,000
Ratio of above	0.235
Triplex safety valve, diameter, inches	3½
Boiler stop valve, diameter, inches	5½
Feed pipes, diameter, inches	2½
Per cent. of air space in grates	40
Volume of furnace for one boiler, cubic feet	300

DATA FROM FOUR-HOUR TRIAL.

I.h.p. per square foot of G.S., main engines	16.6
and auxiliaries	17.1
H.S., main engines	0.374
and auxiliaries	0.387
Square feet of heating surface per I.h.p., main engines	2.67
and auxiliaries	2.58

MAIN AND AUXILIARY STEAM PIPING.

In the boiler rooms the main and auxiliary steam pipes are suspended from the protective deck, on the outboard side, and are parallel to the center-line bulkhead. Steam from the boilers is to the auxiliary line and from this to the main steam line through cross connecting valves. The

auxiliary line is cross connected in the forward fireroom and from it, on the starboard side, is an extension to the ice machine and windlass engine. A reducing valve is placed at each machine. The main steam line begins at the after bulkhead of the forward fireroom. At this point it is 8 inches in diameter and increases successively 11, 13 and, finally, to 15 inches. There are extensions from the auxiliary line to pumps, blower engines, ash-hoist engines, sea chests, fire extinguishers, dynamo rooms, in the forward fireroom on the starboard side to the whistle, and on the port side to the heating system. Provision is also made for boiler-tube cleaners in each fireroom. Expansion joints are fitted where needed throughout the systems. The boiler stop valves are operated by a steam-closing safety device. On the spindle of each stop valve is a piston to which steam may be applied, to close the valve, through a stop valve on the other side of the center-line bulkhead.

In the engine rooms the main steam lines are cross connected, and the steam passes through a separator near the throttle valves of the main engines. The main steam stop valves are designed to be closed by steam in the same manner as the boiler stop valves. The main steam pipe has the following extensions: to auxiliary steam line, bleeder to main condenser, and steam to receivers. The auxiliary steam passes through a separator on the forward engine-room bulkhead to a distributing casting from which are extensions to: jackets of main engines, main circulating pump, auxiliary condenser pump, turning and reversing engines, main air pump, boiling-out condensers, sea chests and to a distributing casting in after end of engine room on the center-line bulkhead, where the line is cross connected. Extensions from this casting are to the main feed and feed supply pumps, on the port side to the heating system, and on the starboard side, through a reducing valve in the engine room and a separator in the steering-engine room, to the steering engine.

EXHAUST PIPING.

In the firerooms forward the auxiliary line is cross connected. It is parallel to the auxiliary steam line through the firerooms, and on the port side, forward, it receives the exhaust from the ice machine and windlass engine, and through the firerooms the exhaust from all auxiliary machinery, heating system and dynamo rooms. There is an extension to the after escape pipe on the starboard side.

In the engine rooms the main exhaust is direct to the main condensers. The auxiliary line is cross connected at the after end of the engine room. It receives on the port side the exhaust from the heating system and steering engine, on the starboard side the vapor from evaporators, and the exhaust from all auxiliary machinery.

The auxiliary line is led to the auxiliary and main condensers through adjustable spring back-pressure valves, to the feed-water heater, and to the L.P. receivers.

The auxiliary condenser is mounted on a combined air and circulating pump. The pump is the horizontal simplex, piston type of Blake design, fitted with medium rubber valves and piston rods of rolled Tobin bronze.

DATA FOR AUXILIARY CONDENSER.

Number per ship	2
Tubes, number	608
B.W.G., number	16
outside diameter, inch	¾
cooling surface, square feet	614
length between tube sheets, inches	74
Shell, inside diameter, inches	28
thickness, inch	¼
Nozzle, exhaust, diameter, inches	9
air-pipe suction, diameter, inches	5
injection, diameter, inches	5
discharge, diameter, inches	5

DATA FOR PUMP.

Cylinder, steam, diameter, inches	6
air, diameter, inches	10
water, diameter, inches	10
Stroke, inches	10

Piston rod, diameter, inches	1¼
Nozzle, steam, diameter, inch	1
exhaust, diameter, inch	1¼
water suction, diameter, inches	5
discharge, diameter, inches	3
air suction, diameter, inches	6
discharge, diameter, inches	5
Ratio of combined area through all suction valves to mean area of piston :	
Air	0.557
Water	0.557

STEAM PIPE DRAINS.

In the firerooms the main and auxiliary steam lines drain, through traps, to pipes, supported from the outboard bulkheads, which lead to the engine rooms.

In the engine rooms the drains from the main and auxiliary steam lines and from evaporator coils join the drain pipes from the firerooms and are led to the main feed tanks, and main or auxiliary condensers. The feed-water heaters drain direct to the main or auxiliary condensers.

The steering engine and dynamo separators drain through traps to the exhaust line.

The whistle steam pipe has a drain near its lower end which unites with the drain from a separator immediately below the whistle and discharges through a trap to the fireroom drain pipe. The ice machine and windlass engines drain direct to the exhaust line. Automatic "float traps" are used throughout, and they are all arranged to be bypassed.

MAIN CIRCULATING PUMP.

This pump is forward of the main condenser and is the centrifugal, double-inlet type, driven by a direct-connected compound engine. There are two 15½-inch main injection valves, one at a higher level than the other; one 16-inch main drain and one 21½-inch bilge injection valve so arranged by an interlocking system that only one valve may be opened at a time to the suction of this pump. A pipe

to by-pass the condenser is installed for the pump to discharge directly overboard. The impeller is not shrouded. This and the casings are composition castings.

DATA FOR CIRCULATING PUMP.

Number per ship	2
Diameter, H.P. cylinder, inches	11
L.P. cylinder, inches	19
Piston rod, inches	2¼
Ratio of piston areas	2.98
Stroke, inches	12
Connecting rod, length between centers, inches	27
diameter, inches	2¼ and 2¾
Crank shaft, diameter, inches	5¼
length of bearings, inches	6½, 10, and 10
crank pin, diameter, inches	5½
length, inches	6½
Crank angle, degrees	90
Crosshead, area, square inches	74
pin, diameter, inches	4
length, inches	5½
Width of piston-packing ring, radially, inch	⅝
axially, inches	1⅛
Valve travel, inches	3½
H.P., angular advance, degrees	35
L.P., angular advance, degrees	37½
diameter, H.P., inches	4
L.P., inches	9

	Top.	*Bottom.*
Valve cut-off per cent. of stroke, H.P.	75	69
L.P.	74	67
steam lead, H.P., inch	¼	3/16
L.P., inch	3/16	¼
port areas, H.P., steam, square inches	9.0	9.7
exhaust, square inches	11.6	11.6
L.P., steam, square inches	20.4	21.8
exhaust, square inches	24.6	24.6
Nozzle, H.P., steam, diameter, inches	3	.....
exhaust, diameter, inches	4½	.....
L.P., steam, diameter, inches	4½	.....
exhaust, diameter, inches	5½	.....
Pump suction nozzle, diameter, inches	21	.....
discharge nozzle, diameter, inches	21	.....
impeller, diameter, inches	54	.....
width of tip of blade, inches	5¼	.....

TEST OF MAIN CIRCULATING PUMP.

The test on this pump was to determine the relation between the revolutions per minute and the corresponding indicated horsepower developed. The suction and discharge nozzles were tapped for gauges to determine the pressures at these points. Both main injection valves were open, and the pump discharged through the condenser as in service condition generally.

It was desired to introduce a Pitot tube into the suction of the pump to determine the volume of water discharged per minute, but the length of straight pipe is too short to eliminate eddy effects, so that data would perhaps not be reliable. The capacity of the pump when drawing from the bilge and discharging directly overboard is about 18,000 gallons when the pump is making 250 revolutions per minute.

DATA OF TEST.

Time	3'11	3'30	4'35	4'50
Suction gauge reading	4 in.	1 lb.	3 lbs.	4 lbs.
Discharge gauge reading, pounds	17	12	10	8
Total head, pounds	19	11	7	4
Exhaust gauge reading, inches	23	18	17	17
Steam gauge reading	205	220	215	215
Revolutions per minute	250	200	156	105
Indicator spring:				
H.P., number	100	100	80	80
L.P., number	30	30	16	16
Combined area, square inches:				
Card No. 1, H.P.	8.15	4.80	3.99	2.15
L.P.	7.83	2.70	4.12	1.92
Card No. 2, H.P.	.....	4.40	4.02	2.08
L.P.	.....	2.85	4.31	1.88
Combined M.E.P.:				
Card No. 1, H.P.	218.0	125.0	83.2	44.8
L.P.	61.7	22.1	18.1	8.52
Card No. 2, H.P.	.....	114.0	84.0	43.4
L.P.	.....	22.8	19.0	8.32
Mean I.H.P.:				
H.P.	152	67.0	36.4	12.9
L.P.	127	38.2	24.7	7.5
Total	279	105.0	61.1	20.4

Steam pressure in valve chest, pounds, gauge:				
Card No. 1	180	120	74	34
2	.....	115	74	34
Mean	180	118	74	34
Piston speed, feet, per minute	500	400	312	210
Piston displacement, cubic feet, per minute:				
H.P.	320	256	199	134
L.P.	975	780	608	410
Ratio	3.05	.....	.....	.....
Velocity through ports, in feet, per minute:				
H.P. steam, top	5,120	4,090	3,180	2,140
bottom	4,770	3,820	2,970	2,000
exhaust.	3,970	3,180	2,470	1,660
L.P. steam, top	6,900	5,510	4,300	2,900
bottom	6,450	5,160	4,020	2,710
exhaust	5,700	4,560	3,560	2,400
Velocity through pipes, in feet, per minute:				
Steam	6,520	5,220	4,060	2,740
Exhaust	5,910	4,730	3,690	2,480

DATA FROM STANDARDIZATION TRIAL.

Run number.	Revolutions per minute.		Indicated horsepower.	
	Port.	*Starboard.*	*Port.*	*Starboard.*
1 to 5	165	153	62	53
6	168	170	66	68
7	144	146	45	46
8	120	120	29	29
9	110	126	24	33
10	130	108	36	22
11	108	100	22	19
12	105	100	21	19
13	100	106	19	22
14	100	42	19	4
Four-hour trial.	173	174	71	72

MAIN CONDENSER.

There is one main condenser, with tubes horizontal, on the outboard side of each engine room. The shell is steel plate stiffened with angles, T-shapes and tie rods. The sides are vertical and parallel, with the top and bottom bent to a radius of 32½ inches. This was necessary be-

cause the circular shape would take up too much space. The tube sheets are rolled Muntz metal, and the condenser heads composition. The condenser tubes are seamless-drawn and made from the Government formula.

Baffle plates are placed beneath the main exhaust nozzle, and lower down near the ends of the tubes is a second set of baffles. The condenser is clothed with hair felt and lagged with Russia sheet iron.

DATA FOR CONDENSER.

Number per ship	2
Tubes, number	6,169
B.W.G., number	16
outside diameter, inch	$\frac{5}{8}$
length between tube sheets, inches	174
cooling surface, square feet	14,640
Number of supporting plates	2
Thickness of shell plate, inch	$\frac{5}{16}$
tube sheet, inch	1
Diameter of relief valve, inches	3
Nozzles, main exhaust, diameter, each, inches	28
auxiliary exhaust, diameter, inches	9
injection and discharge, diameter, each, inches	21
main air pump, diameter, inches	13
manhole, diameter, each, inches	13

DATA FROM STANDARDIZATION TRIAL.

Mean temperature, degrees Fahrenheit:	*Port.*	*Starboard.*
Injection	35	35
Discharge, overboard, runs 1 to 5	107	111
runs 6 to 14	93	104
of air pump, runs 1 to 5	61	64
runs 6 to 14	61	52
Vacuum, inches	26.4	26.6

DATA FROM FOUR-HOUR TRIAL.

Mean temperature, degrees Fahrenheit:		
Injection	42	42
Discharge, overboard	109	112
of air pump	70	63
Vacuum, inches	25.9	26
Square feet of cooling surface per indicated horsepower of main engines	1.13	1.08

MAIN AIR PUMP.

The main air pump is aft of the main condenser, and the suction chamber is above the level of the bottom of the condenser. This causes the pump to operate a little unsteadily. A curve of performance was not determined for this pump while the ship was at the dock, because a satisfactory load could not be given it. Hence it was decided to wait until the standardization trial and obtain the indicated horsepower in relation to the revolutions per minute of the main engines. This could not be satisfactorily done and is omitted. The pump is a Blake vertical, twin beam, and discharges to the main feed tank. The valves are phosphor-bronze discs, held against their seats by springs.

DATA FOR MAIN AIR PUMP.

Number per ship	2
Cylinders, steam, diameter, inches	14
water, diameter, inches	35
Piston rod, steam, diameter, inches	2¼
water, diameter, inches	3¼
Stroke, inches	18
Nozzle, steam, diameter, inches	2
exhaust, diameter, inches	2½
suction, diameter, inches	13
discharge, diameter, inches	11
Per cent. area of valves to piston area, foot	24
head	24
piston	24

DATA FROM FOUR-HOUR TRIAL.

	Port.	*Starboard.*
Mean double strokes per minute	20.2	20.7
vacuum, in inches	25.9	26.0
indicated horsepower	21.0	21.0
Per cent. of I.h.p. of main engines	0.15	0.16
Piston speed, in feet per minute	61	
Piston displacement, in cubic feet per minute :		
2 air pump cylinders	405	
2 steam cylinders	128	
2 L.P. pistons of main engines	60,800	
Displacement ratio $\frac{\text{L.P. piston}}{\text{Air pump}}$	150	
Velocity, steam, in feet, per minute	5,880	
exhaust, in feet, per minute	3,750	
suction, in feet, per minute	440	
discharge, in feet, per minute	614	

FEED SYSTEM.

The main feed tanks are in the after outboard side of each engine room. All steam trap drains discharge—except those of the heating system—to the bottom of this tank through a device which operates similarly to an injector, and is noiseless in operation. The water is drawn from this tank by a feed-supply pump and discharged through a feed heater to the suction of the main feed pumps, which are in the engine rooms. The discharge of the main feed pumps—cross connected in the engine room by a valve on the port side—extends forward through the firerooms—one section on the starboard side and one on the port side—with extensions to boilers. The pumps and feed-water heaters are on the center-line bulkhead, the feed-water heaters forward.

The auxiliary feed pumps are in the firerooms on the center-line bulkhead. The suction is from the main or reserve feed tanks and may be from the discharge of the feed-supply pumps through the feed-water heaters. The auxiliary feed line is not cross connected, but extends fore and aft with extensions from each pump and to boilers.

FEED AND FILTER TANK.

This tank is constructed of galvanized steel, clothed with hair felt and lagged with galvanized steel plates. The upper part is divided by division plates for filtering. "Loofa," the filtering material, is placed in each compartment. The water passes down through the "loofa" to the top of the succeeding compartment three times before passing into the feed tank. The following discharge to the filter compartment: main air pump, auxiliary and dynamo condenser pumps, feed supply pump, and fresh water from the evaporating plant. The reserve-feed tanks are in the inner bottom.

DATA OF FEED-WATER VOLUMES.

	Cubic feet.
Feed and Filter Tank :	
Port	700
Starboard	700
Compartment B-92	547
B-93	547
B-94	775
B-95	775
B-96	652
B-97	652
B-98	753
B-99	753
Total	6,850

FEED HEATER.

The heater is horizontal and supported by brackets from the center-line bulkhead. The auxiliary exhaust passes through a regulating valve on the top, and drains to the main and auxiliary condensers. The feed water passes through the tubes.

DATA FOR FEED HEATER.

Number per ship	2
Tubes, number	857
B.W.G., number	18
outside diameter, inch	$\frac{5}{8}$
heating surface, square feet	980
length between tube sheets, inches	84
Shell, inside diameter, inches	$38\frac{1}{8}$
thickness, inch	$\frac{5}{16}$
Diameter of nozzles, auxiliary exhaust, inches	9
drain, inches	$2\frac{1}{2}$
water inlet and outlet	$6\frac{1}{2}$

DATA FROM STANDARDIZATION TRIAL.

Mean temperature of water, degrees Fahrenheit:	*Port.*	*Starboard.*
To feed heater	60.9	56.4
From feed heater	211	179
Mean auxiliary exhaust pressure, during highest speed runs, pounds, gauge	20	

DATA FROM FOUR-HOUR TRIAL.

Mean temperature of water, degrees Fahrenheit :		
To feed heater	70	63
From feed heater	198	*121
Mean auxiliary exhaust pressure, pounds, gauge	22	
Maximum observed temperature, degrees Fahrenheit,	228	222

* The starboard feed water heater-head broke after 1½ hours and it was by-passed for the remainder of the trial.

PUMPS.

Data of Pumps.	Feed supply.	Main feed.	Auxiliary feed.	Fire and bilge.
Number per ship	2	4	8	2
Diameter, cylinder, steam, inches	12	14	11	12
water, inches	16	10	8	10
pipe, steam, inches	1½	2	1½	1½
exhaust, inches	2	2½	1½	2
suction, inches	6½	5	4½	5
discharge, inches	6½	5	4	5
Piston rod, diameter, inches	2½	2¾	2½	2½
Stroke, inches	16	18	12	12
Type	vertical, simplex, piston.	vertical, simplex, plunger, center-packed.	vertical, simplex, plunger, center-packed.	vertical, simplex, piston.
Valves	hard rubber.	brass check.	brass check.	medium rubber.

All pumps are Blake design.

Material for all water piston rods: rolled Tobin bronze.

Ratio of combined area through all suction valves to mean area of water piston:

Feed supply	0.503
Main feed	0.705
Auxiliary feed	0.672
Fire and bilge	0.606

Manifold valves of pumps.	Suction.	Discharge.
Feed supply	Main feed tank. Reserve feed tank. Water-boat filling pipe. Air-pump channel-way.	Main feed tank. Feed-water heater. Main feed-pump suction. Auxiliary feed-pump suct. Reserve feed tanks.
Main feed	Feed-water heater. Main feed tanks.	Main feed system.
Auxiliary feed	Feed-water heater. Main feed tank. Pumping-out boilers. Bilge drainage. Sea. Hose connection. C. & R. manifold.	Auxiliary feed-system. Fire main. Overboard. Hose connection.
Fire and Bilge	Sea. Bilge drainage. Crank pits. C. & R. manifold.	Overboard. Fire main. Distillers.

TEST OF PUMPS.

The tests of the pumps were made to obtain a curve which would indicate the relation between the strokes per minute and the corresponding indicated horsepower developed. The discharge pressure was maintained constant throughout the test, and by adjusting the relief valve the volume of discharge was regulated. The strokes were varied by throttling the steam supply and adjusting the discharge relief valve. One pump of each type only was tested. Two indicator cards were taken for each condition.

The curve thus obtained is accurate only for the condition of the test, and to make it of general application an interpolation formula has been derived. The probable error in this formula is perhaps no greater than the variation between pumps of the same type and the reading of the discharge gauge. This formula is based on the supposition that the length of stroke will be constant, for a given number of strokes per minute, and also the friction of the pump. The curves are drawn to make angles as nearly equal as possible with each coördinate to reduce the error of reading to a minimum.

The maximum piston speed was as near 100 feet per minute as the pump would operate satisfactorily. No attempt was made to compute the steam consumption from the indicator cards. Experiments made in the Steam Engineering Laboratories of Drexel Institute by the writer during evening-class work for the past three years indicate that the ratio between the actual steam consumption and that computed from the indicator card will vary with the type of engine and the condition under which it may be operating. It was impossible to determine this ratio for these pumps at the time the tests were made.

INTERPOLATION FORMULA.

Symbols:

I.H.P. = Indicated horsepower desired.

I.h.p. = Indicated horsepower as determined from curve of test.

W.h.p. = Water horsepower corresponding to a given I.h.p.

C = Constant.

c = Constant.

K = Constant.

N = Double strokes per minute observed for any condition.

A = Area of water piston.

S = Stroke in feet.

p = Discharge gauge reading for test.

$k = \frac{1}{p} = \text{constant.}$

P = Discharge pressure observed for any condition.

Condition of test:

$$\text{I.h.p.} = C \times \text{W.h.p.}$$
$$= C \times \frac{2\,N\,A\,S}{33{,}000}\,p.$$
$$= C \times c \times p.$$
$$= Kp.$$

$$\text{I.H.P.} \propto P.$$

Therefore, $\text{I.H.P.} = \text{I.h.p.} \times \frac{P}{p} = k \times \text{I.h.p.} \times P.$

The I.h.p. corresponding to the observed number of double strokes per minute is determined from the curve and substituted in the interpolation formula.

The following are the tests and the curves of performance derived from them:

AUXILIARY FEED PUMP U. S. A. C. *TENNESSEE.*

DATA OF TEST.

Discharge gauge reading 275 pounds.

Time	2'09	2'17	2'30	2'40
Double strokes, per minute	38	29	21	9.5
Discharge gauge reading	275	275	275	275
Steam gauge reading	260	260	260	255
Exhaust gauge reading, inches	11	11	12	12
Indicator spring, number	150	150	150	150
Area of card, square inches:				
No. 1, top	3.45	3.13	3.52	3.48
bottom	3.55	3.14	3.45	3.42
No. 2, top	3.40	3.14	3.63	3.37
bottom	3.53	3.07	3.70	3.23
Stroke in feet	1.00	0.98	0.96	0.91

Mean M.E.P.:				
Card No. 1	153	155	160	167
2	152	155	168	160
Mean	153	155	164	164
Theoretical:				
Card No. 1	202	183	188	187
2	202	178	195	184
Mean	202	181	192	186
Ratio $\frac{\text{Actual}}{\text{Theoretical}}$	0.76	0.85	0.86	0.88
Steam pressure in valve chest, pounds, gauge:				
Card No. 1	218	185	188	183
2	218	180	195	180
Mean	218	183	192	182
I.H.P.:				
Card No. 1	32.8	25.4	18.2	8.15
2	32.6	25.4	19.1	7.80
Mean	32.7	25.4	18.7	7.98
Piston displacement, cubic feet, per minute:				
Steam	49.1	36.8	26.1	11.2
Water	25.5	19.0	13.5	5.8
Piston speed, feet, per minute	76.0	56.8	40.3	17.3
Velocity through nozzle, in feet per minute:				
Suction	231	172	122	52.6
Discharge	292	217	154	66.4
Steam	5,740	4,310	3,050	1,310
Exhaust	3,990	2,990	2,120	914

FEED SUPPLY PUMP U. S. A. C. *TENNESSEE.*

DATA OF TEST.

Discharge gauge reading 22 pounds.

Time	2'41	2'52	3'03	3'20	3'38
Double strokes, per minute	48	40	30	19	9
Discharge gauge reading	24	22	22	22	22
Steam gauge reading	185	154	156	175	170
Exhaust gauge reading, in	16	15	14	17	15
Indicator spring, number	100	100	100	40	40
Area of card, square inches:					
No. 1, top	2.85	2.28	1.64	3.15	2.84
bottom	2.85	2.31	1.58	3.10	2.92
No. 2, top	2.80	2.25	1.65	3.08	2.86
bottom	2.87	2.29	1.59	3.09	2.86
Stroke, in feet	1.33	1.33	1.29	1.23	1.16

Mean M.E.P.:					
Card No. 1	80.5	62.0	46.7	37.9	37.3
2	80.0	63.0	46.9	37.6	37.0
Mean	80.3	62.5	46.8	37.8	37.2
Theoretical:					
Card No. 1	106	81	55	43	40
2	104	80	56	42	41
Mean	105	81	56	43	41
Ratio $\frac{\text{Actual}}{\text{Theoretical}}$	0.76	0.77	0.84	0.88	0.91
Steam pressure in valve chest, pounds, gauge:					
Card No. 1	115	81	50	36	35
2	113	80	50	35	34
Mean	114	81	50	36	35
I.H.P.:					
Card No. 1	34.7	22.3	12.5	6.47	3.10
2	34.5	22.6	12.6	6.42	2.99
Mean	34.6	22.5	12.6	6.45	3.05
Piston displacement, cubic feet, per minute:					
Steam	98.5	82.1	59.7	36.1	16.1
Water	177	148	107	64.6	28.9
Piston speed, feet per min.	128	106	77.4	46.8	20.9
Velocity through nozzle, in feet, per minute:					
Suction	770	643	465	281	125
Discharge	770	643	465	281	125
Steam	8,000	6,670	4,850	2,940	1,310
Exhaust	4,520	3,760	2,740	1,650	740

MAIN FEED PUMP U. S. A. C. *TENNESSEE*.

DATA OF TEST.

Discharge gauge reading 275 pounds.

Time	10'55	11'06	11'16	11'25
Double strokes, per minute	27	21	4.5	11
Discharge gauge reading	275	275	275	275
Suction gauge reading	22	22	22	22
Steam gauge reading	255	265	260	260
Exhaust gauge reading, inches	11	12	12	12
Indicator spring, number	150	150	150	150
Area of card, square inches:				
No. 1, top	3.30	3.12	3.00	3.00
bottom	3.20	3.24	2.80	2.98
No. 2, top	3.15	3.05	3.09	3.05
bottom	3.20	3.17	3.03	2.89
Stroke, in feet	1.50	1.46	1.42	1.41

Mean M.E.P.:				
Card No. 1	150	150	140	146
2	145	147	147	145
Mean	148	149	144	146
Theoretical:				
Card No. 1	174	172	160	156
2	177	170	162	156
Mean	176	171	161	156
Ratio $\frac{\text{Actual}}{\text{Theoretical}}$	0.84	0.87	0.90	0.94
Steam pressure in valve chest, pounds, gauge:				
Card No. 1	174	170	158	153
2	177	165	159	153
Mean	176	168	159	153
I.H.P.:				
Card No. 1	55.7	42.1	26.4	20.7
2	53.8	41.2	27.7	20.6
Mean	53.8	41.7	27.1	20.7
Piston displacement, in cubic feet, per minute:				
Steam	84.3	64.0	42.8	32.4
Water	42.5	32.2	21.5	16.3
Piston speed, feet, per minute	81.0	61.4	41.2	31.1
Velocity through nozzle, in feet, per minute:				
Suction	312	237	158	120
Discharge	312	237	158	120
Steam	3,870	2,940	1,970	1,490
Exhaust	2,480	1,880	1,260	952

FIRE AND BILGE PUMP, U. S. A. C. *TENNESSEE.*

DATA OF TEST.

Discharge directly overboard.

Time	3'50	3'58	4 09
Double strokes per minute	45	34	24
Discharge gauge reading	Discharge overboard.		
Steam gauge reading	260	260	265
Exhaust gauge reading, inches	12	12	12
Indicator spring, number	40	40	20
Area of card, square inches:			
No. 1, top	1.64	1.34	2.36
bottom	1.50	1.28	2.31
No. 2, top	1.62	1.38	2.39
bottom	1.50	1.25	2.34
Stroke, in feet	1.00	0.97	0.95

Mean M.E.P.:			
Card No. 1	18.9	16.0	14.6
2	18.3	16.0	14.8
Mean	18.6	16.0	14.7
Theoretical:			
Card No. 1	27	23	18
2	27	23	18
Mean	27	23	18
Ratio $\frac{\text{Actual}}{\text{Theoretical}}$	0.69	0.70	0.82
Steam pressure in valve chest, pounds, gauge:			
Card No. 1	20	16	10
2	20	16	10
Mean	20	16	10
I.H.P.:			
Card No. 1	5.73	3.56	2.24
2	5.54	3.56	2.27
Mean	5.64	3.56	2.26
Piston displacement, cubic feet, per minute:			
Steam	69.4	51.0	35.2
Water	47.8	35.1	24.2
Piston speed, feet, per minute	90.0	66.0	45.6
Velocity through nozzle, in feet, per minute:			
Suction	352	258	178
Discharge	352	258	178
Steam	5,650	4,150	2,870
Exhaust	3,180	2,340	1,610

FIRE AND BILGE PUMP, U. S. A. C. *TENNESSEE.*

DATA OF TEST.

Discharge gauge reading **100** pounds.

Time	2'33	2'41	2'46	3'00	3'12
Double strokes per minute	53	40	30	21	10
Discharge gauge reading	95	100	100	100	100
Steam gauge reading	255	250	255	265	255
Exhaust gauge reading, inches	16	16	16	16	16
Indicator spring, number	100	100	100	40	40
Area of card, square inches:					
No. 1, top	3.12	2.83	2.68	5.81	5.35
bottom	2.98	2.78	2.61	5.75	5.67
No. 2, top	3.22	2.79	2.70	5.81	5.36
bottom	3.03	2.72	2.67	5.91	5.64
Stroke, in feet	0.983	1.00	1.00	0.986	0.955

Mean M.E.P.:					
Card No. 1	87.2	79.1	74.8	66.1	64.8
2	90.0	77.6	75.2	66.5	65.1
Mean	88.6	78.4	75.0	66.3	65.0
Theoretical:					
Card No. 1	124	113	97	76	75
2	127	102	93	79	74
Mean	126	108	95	78	75
Ratio $\frac{\text{Actual}}{\text{Theoretical}}$	0.70	0.73	0.79	0.85	0.87
Steam pressure in valve chest, pounds, gauge:					
Card No. 1	133	111	91	70	68
2	138	98	86	73	66
Mean	136	105	89	72	67
I.H.P.:					
Card No. 1	30.6	21.3	15.1	9.22	4.18
2	31.6	20.9	15.2	9.27	4.19
Mean	31.1	21.1	15.2	9.25	4.19
Piston displacement, cubic feet, per minute:					
Steam	80.5	61.8	46.4	32.0	14.8
Water	55.3	42.4	31.8	22.0	10.1
Piston speed, feet, per minute	104.0	80.0	60.0	41.4	19.1
Velocity through nozzle, in feet, per minute:					
Suction	407	312	234	162	74.3
Discharge	407	312	234	162	74.3
Steam	6,550	5,000	3,780	2,600	1,200
Exhaust	3,700	2,840	2,130	1,470	678

DATA FROM TRIALS.

The auxiliary feed pumps were not used. The main feed pumps operated somewhat unsteadily, at times, perhaps, due to the formation of vapor below the pistons. The mean values only, for port and starboard engine rooms, for each condition, are given. The fire and bilge pumps discharged overboard from bilge.

Abbreviations: F.S. = Feed supply pump.
M.F. = Main feed pump.
F.& B. = Fire and bilge pump.

Run No.	Double strokes per minute.			Total water piston displacement, in cubic feet, per minute.		
	F.S.	M.F.	F.& B.	F.S	M.F.	F.& B.
1 to 5	23	26	47	160	164	100
6	23	23	52	160	152	108
7	22	21	54	152	128	112
8	20	19	58	138	116	120
9	19	17	59	130	100	122
10	19	24	62	130	148	128
11	12	14	60	78	84	124
12	10	14	59	64	84	122
13	11	12	61	72	72	126
14	13	10	64	86	60	130
4-hour trial	25	27	44	176	172	94

Run No.	Discharge gauge reading.		Total indicated horsepower.		
	F.S.	M.F.	F.S.	M.F.	F.& B.
1 to 5	22	346	17	262	12.1
6	22	346	17	227	14.5
7	22	346	15	207	15.5
8	22	346	13	187	17.6
9	22	346	13	166	18.2
10	22	346	13	242	19.8
11	22	346	8	136	18.8
12	22	346	7	136	18.2
13	22	346	7	116	19.4
14	22	346	9	96	22.0
4-hour trial	22	340	19	267	10.8

BLOWERS.

The fireroom blowers, manufactured by the B. F. Sturtevant Company, are in the firerooms suspended from the berth deck. There are two for each fireroom, arranged with one inboard and one outboard. The suction for the blowers is from the fireroom ventilators, with a device for closing the suction of each blower. As installed the crank shaft is above the cylinders. A water service is installed throughout the firerooms, for the blowers, which is supplied by the distiller circulating pump. The engines are the double enclosed type.

DATA FOR BLOWER.

Number per ship	16
fireroom	2
Cylinders, diameter, inches	5

Cylinders, number	2
stroke, inches	4
Crankshaft, diameter, inches	2 7/8
bearings, inches	7½
crank pin, diameter, inches	3
length	3½
Crank angle, degrees	180
Connecting rod, length between centers, inches	12
diameter, inches	1¾ and 1½
Crosshead pin, length, inches	2¼
diameter, inches	2
slipper, area, square inches	7.9
Piston rod, diameter, inches	1½
Valve travel, inches	2¼
diameter, inches	2½
Diameter of fan, inches	60
Width of tip of blades, inches	18
Area of induction nozzle, square inches	1,000
eduction nozzle, square inches	1,500
Diameter of nozzles, steam, inches	2
exhaust, inches	2½

TEST OF BLOWER.

The method for this test was to close one fireroom as would be done for a forced-draft condition. Two blowers were run at 600 revolutions per minute, and the ash-pan doors adjusted until the fireroom pressure was equal to two inches of water. This adjustment was not changed throughout the test. The revolutions were varied and the corresponding air pressures noted. Inertia effects in the indicator used made it impossible to obtain cards above 400 revolutions per minute, which probably makes the indicated horsepower at this condition in excess of its true value. The I.H.P. curve is extended beyond this point, although there is no exact data for it.

The cylinder next to the valve is designated "aft cylinder," and the other cylinder "forward cylinder."

DATA OF TEST.

Time	3'35	4'08	5'11	2'25
Revolutions per minute	300	400	200	600
Air pressure, inches of water	0.6	1.1	0.1	2.0
Steam gauge reading	165	158	160	165

Exhaust gauge reading, inches	15	15	15	15
Indicator spring, number	100	100	60 and 40	
Area of card, square inches:				
No. 1, forward cylinder	0.70	1.55	0.55	
aft cylinder	0.70	1.47	0.32	
No. 2, forward cylinder	0.70	1.43	0.54	
aft cylinder	0.70	1.50	0.35	
Mean M.E.P.:				
No. 1, forward cylinder	28.6	48.4	9.17	
aft cylinder	28.6	55.7	8.01	
No. 2, forward cylinder	28.7	46.2	9.01	
aft cylinder	28.5	58.8	8.75	
Steam press. in valve chest, lbs., gauge.	12	34	—3	
I.H.P.:				
No. 1, forward cylinder	3.31	7.47	0.707	
aft cylinder	3.31	8.60	0.618	
No. 2, forward cylinder	3.32	7.11	0.695	
aft cylinder	3.30	9.08	0.675	
Total I.H.P.:				
Card No. 1	6.62	16.1	1.33	
2	6.62	16.2	1.37	
Mean	6.62	16.2	1.35	
Piston displacement, for both pistons, cubic feet, per minute	53.0	70.6	35.3	106.0
Piston speed, feet, per minute	200.0	266.0	133.0	400.0
Velocity through nozzle, in feet, per minute:				
Steam	2,430	3,240	1,620	4,860
Exhaust	1,550	2,070	1,040	3,110

DATA FROM STANDARDIZATION TRIAL.

Run number.	Indicated horsepower.		Air pressure inches of water.		Revolutions per minute.	
	One.	Total.	Port.	Starboard.	Port.	Starboard.
N-1	27.8	445	2.0	2.0	545	600
S-2	27.6	444	1.9	1.9	543	609
N-3	27.5	400	1.9	1.9	536	610
S-4	27.9	446	1 9	1.9	548	614
N-5	23.2	371	2.0	2.0	552	504
S-6	8.0	128	0.65	0.65	350	308
N-7	5.0	80	0.34	0.34	242	302
S-8	.....	.....	0.10	0.10	244	270
N-9	.....	.....	0.11	0.11	257	239
S-10	.....	.....	0.13	0.13	276	207
N-11	.....	.....	.....	.....	254	175
S-12	.....	.....	.....	.....	238	156
N-13	.....	.....	.....	.....	232	.....
S-14	.....	.....	.....	.....	.....	.....

DATA FROM FOUR-HOUR TRIAL.

Time.	Air pressure, inches of water.		Revolutions per minute.	
	Port.	Starboard.	Port.	Starboard.
Start	.....	.....	497	501
Hour No. 1	1.57	1.57	510	504
2	1.45	1.45	432	485
3	1.38	1.38	459	470
4	1.38	1.38	.....	475
Mean from official report of observations taken every thirty minutes	1.47	1.47	484	488
Mean indicated horsepower from curve :				
One blower			19.8	20.2
Eight blowers			157.4	161.6
Total			319	

ASH HOISTS.

The ash-hoist engines, of the contractor's design, are placed in the smoke-pipe hatches near the fireroom ventilator shafts. In these shafts are guides for the ash buckets, and openings on the main deck for operating the engines and discharging ashes. In the second and aft fireroom ventilator shafts, port side only, there are additional openings on the boat deck for supplying coal to the launches. Suspended from the boat deck are trolley tracks from the openings in the ventilator shafts to the ash chutes, of which there are three on each broadside. The engines will raise a weight of 300 pounds from the fireroom floors to the main deck in five seconds with 80 pounds of steam. The engines are fitted with a reversing and follow-up gear, and a safety device to stop the engine when the bucket reaches either end of its travel.

DATA FOR ONE ENGINE.

Number, per ship	8
Cylinders, per engine	2
diameter, inches	4½
stroke, inches	4½
Piston rod, diameter, inch	1
Type of valve	Piston.
Valve travel, inches	1½
diameter, inches	2½
steam and exhaust lap, inch	1/8
angular advance, degrees	90

Connecting rod, length between centers, inches	10
diameters, inch	$\frac{7}{8}$ and 1
Crosshead slipper, area, square inches	11
pin, diameter, inch	1
length, inches	$1\frac{1}{2}$
Crank shaft, diameter, inches	2
crank pin, diameter, inches	$1\frac{3}{4}$
length, inches	$1\frac{1}{2}$
angle, degrees	90
length of bearing, inches	$3\frac{1}{2}$
Whip, drum, diameter, inches	5
length, inches	10
steel, diameter, inch	$\frac{3}{8}$
Pipes, diameter, inches:	
Steam	$1\frac{1}{2}$
Exhaust	$1\frac{1}{2}$

MACHINE SHOP.

The machine shop is on the berth deck between the engine-room hatches. The entrances are direct from the engine rooms. The following machines are installed:

1 30-inch extension gap lathe, to swing 15 inches over upper ways, and length between centers when extended about 11 feet.

1 14-inch screw-cutting back-geared lathe, to have an 8-foot bed.

1 Shaping machine, about 15-inch stroke and 26-inch traverse.

1 Back-geared vertical drill press.

1 16-inch sensitive drill.

1 Vertical milling machine.

1 Emery grinder, with two 12-inch wheels.

1 30-inch grindstone.

1 Combined hand punch and shears, to be installed in the blacksmith shop.

These machines are furnished with the requisite tools and attachments. Lockers are provided where needed, and for the ship's indicators. One work bench is fitted in after end of shop. A motor is installed to drive the tools. This was tested to determine the power necessary to op-

erate all machines at as heavy a load as could be given them under the circumstances. A piece of steel or cast iron was put in each machine and the tool adjusted to its heaviest cut. A piece of steel was ground by the grindstone and emery wheels only when the observations on the volt and ammeter were to be made. Three tests were made: one with all the machines in operation, one with the machines in operation but tools not cutting, and the last to determine the friction of the shafting. The motor is rated by the builders at 15 electrical horsepower.

DATA OF TEST.	Electrical horsepower. Input of motor.
To operate all tools, cutting	3.78
not cutting	2.72
shafting	1.62
Power absorbed :—	
By shafting	1.62
tools, friction	1.10
cutting tools	1.06

EVAPORATING PLANT.

The evaporating plant is located on the berth deck forward of the engine-room hatch. On this deck are three evaporators; the distiller-circulating, evaporator-feed and fresh-water pumps; the water meter and a small reservoir fresh-water tank. The three distillers are placed vertically in an air trunk. The evaporators are piped for single effect only.

PIPING.

Steam from the auxiliary line is supplied through a reducing valve, in the evaporator room, to a casting with three valves which distribute steam to the evaporators. The vapor from the evaporators is led by a pipe to a manifold on the distillers so that one or more distillers may be used. The discharge from the distiller-circulating pump is also led to a manifold at the distillers, and another manifold receives the circulating water discharged from them. The evaporator feed-pump suction is from the dis-

tiller-circulating discharge water. The steam traps for the evaporator coils discharge to the feed tanks; an expansion trap is used for this purpose for each evaporator. The evaporator bottom blow pipes are led to discharge overboard. There is a connection for the vapor from the evaporators to the auxiliary exhaust line. The distiller circulating pump discharges also to the sanitary system, and there is a connection from the fire and bilge pump to the distiller-circulating water pipe in order that this pump may be used for the distillers.

EVAPORATORS.

The evaporator shells are steel plates; the after head is stiffened by I-beams riveted to it and, in the vapor space, by tie rods to the front head. The evaporator tube headers are composition covered by steel plates. The headers are connected by brass pipes and steel tie rods. The passage for the steam is through all the tubes to the back header, and it returns through the bottom row of tubes to the front header. To the front header are nozzles for steam, drain, relief valve, gauge and air cocks. To the shell are pads for relief valve, vapor pipe, feed and bottom blow valves, salinometer pot, water gauge, and zincs.

DATA FOR EVAPORATOR.

Number per ship	3
Shell, diameter inside, inches	72¾
thickness, inch	⅜
Heads, thickness, inch	9/16
Stays, number	13
diameter, inches	1¼
Tubes, brass, outside diameter, inches	2
B.W.G. number	12
number	128
length between tube sheets, inches	69⅞
square feet of heating surface	390
Diameter, steam nozzle, inches	3
relief valve, coil, inches	2
shell, inches	2
vapor pipe, inches	3½
coil drain pipe, inches	1½
brine blow, inches	1½

DISTILLER.

The distillers are the contractor's design with the tubes expanded into the tube sheets. The upper sheet is permanent and the lower is arranged for thermal changes by bolting to it a casting with a nozzle which passes through a stuffing box in the bottom head of the distiller. Baffle plates constrain the vapor to traverse the tubes three times. The body of the distiller is cast iron, and the tubes are brass, tinned inside and out. The circulating water goes through the tubes.

DATA OF DISTILLER.

Number per ship	3
Body, diameter, inside, inches	14⅝
outside, inches	15¼
Tubes, outside diameter, inch	⅝
B.W.G. number	16
number	153
length between tube sheets, inches	52
square feet of cooling surface	109
Nozzle, vapor, diameter, inches	3½
circulating water, diameter, inches	3½
distilled water, diameter, inches	1½

DATA OF PUMPS.

	Distiller Circulating.	Evaporator Feed.	Fresh Water.
Cylinder, steam, diameter, inches	12	4½	4½
water, diameter, inches	16	5	5
Pipe, steam, diameter, inches	1½	½	½
exhaust, diameter, inches	2	¾	¾
suction, diameter, inches	6	2½	2½
discharge, diameter, inches	5	2	2
Piston rod, diameter, inches	2½	1⅛	1⅛
Stroke, inches	12	6	6
Type	Horizontal, simplex, piston.	Vertical, simplex, piston.	Vertical, simplex, piston.
Valves	Medium rubber.	Medium rubber.	Medium rubber.

All the pumps are Blake design.

Material for all water piston rods: Rolled Tobin bronze.

Ratio of combined area through all suction valves to mean area of water piston:

Distiller circulating pump,..... 0.553
Evaporator feed pump,..... 0.534
Fresh water pump, ... 0.534

TEST OF EVAPORATING PLANT.

This plant is required by the specifications to have a total capacity of 23,000 gallons of water per twenty-four (24) hours. A test was made in Boston Harbor to determine this. The distilled water was tested with a silver-nitrate solution, and after several experiments the evaporators were operated without priming. Perhaps a separator in the vapor pipe near the distillers might assist in supplying them with dry steam, which is absolutely essential to produce fresh water. For this arrangement the vapor valves of the evaporator should be wide open and the shell pressure regulated by the manifold at the distillers; the drain from this separator would be, by gravity, to the evaporators. The vapor velocity calculated for the appended data would indicate that particles of moisture would not drain to the evaporator as at present installed. The distilled water filled alternately two barrels, and a record was kept of the number of times the barrels were filled. For one hour one distiller was used, and after increasing the amount of vapor a second distiller was used to relieve the first, which was apparently loaded a little over its capacity. When the two distillers were in use the distiller circulating water was reduced in order to raise the temperature of the evaporator-feed water, which accounts for the temperature of the distiller discharge and distilled water apparent discrepancy in the appended data.

Perhaps the efficiency of the plant might be increased by heating the evaporator-feed water. The heating agent might be the exhaust steam from the pumps, the drain from the coils, or the brine by keeping the bottom blow continually open. The latter methods would possibly produce the highest temperature for the feed water.

The evaporator-feed pump was in constant operation

and the bottom blow open a little to prevent the density of the brine going above 3/32 hydrometer test, at a temperature of 200 degrees Fahrenheit. The boiling point of salt water at this density is about 216 degrees Fahrenheit. The drain traps for the evaporator coils were open a little throughout the test. The following data were obtained:

	One distiller. 1 hour.	Two distillers. 1 hour.
Time		
Steam reduced, gauge, pounds	75	75
at coils, gauge, pounds	40	42
shell, gauge, pounds	26	26
Distiller circulating pump, double strokes per min.	20	24
Discharge pressure, gauge, pounds	5	5
Temperature in degrees Fahrenheit :		
Injection water	36	36
Distiller discharge and evaporator feed	96	98
Distilled water	66	68
Of room	108	108
Distilled water per 24 hours, gallons	22,900	26,200
Gallons of water evaporated per square foot of heating surface per 24 hours	19.6	22.4
Gallons of water distilled per square foot of cooling surface per 24 hours	210	120
Velocity of vapor through evaporator nozzle, in feet, per minute	6,600	7,600
B.t.u. rejected to distiller by vapor, approximate, per minute	150,000	172,000
Velocity of distilled water through nozzle of distiller, in feet, per minute	170	100
Cubic feet of feed water per minute, based on distilled water with no allowance for blowing down	2.2	2.5

REFRIGERATOR PLANT.

The refrigerator rooms are in two sections, one port and one starboard. They are on the berth deck outboard of the forward barbette, with the ice machine amidships forward of the barbette. An Allen dense-air machine is installed, designed to produce a cooling effect equivalent to the melting of three tons of ice per twenty-four hours.

The piping for the refrigerating air is to the ice tank, which may be by-passed; thence to the refrigerator rooms, and returned through the scuttle butts to the machine.

The rooms are arranged for one entrance compartment to each two rooms. The volumes of the rooms and the cooling surface for each are tabulated below.

DATA OF REFRIGERATOR ROOMS.

	Volume in cubic feet.	Cooling surface, sq. feet.	Cubic feet per square foot of cooling surface.
Port side :			
Room, crews	880	204	4.3
ward, officers	520	130	4.0
Entrance	100	...	...
Starboard side :			
Room, warrant officers	320	130	2.5
junior officers	400	108	3.7
Entrance	85	...	...
Room, Captain	300	111	2.7
Admiral	400	118	3.4
Entrance	80	...	...

DATA OF ICE MACHINE.

Cylinder :	Steam.	Compression.	Expansion.
Diameter, inches	10	9	7½
Stroke, inches	14	14	14
Piston rod, diameter, inches	2	2	2
Cut-off, adjustable, inches	4 to 8	...	...
Crosshead, area, square inches	55	55	55
pin diameter, inches	3	3	3
length, inches	3½	3½	3½
Crank shaft, diameter, inches	5½	5½	5½
bearing, length, inches	10½	...	10½
pin, diameter, inches	4	5½	4
length, inches	4	5½	4
angle, degrees	0	135.8	82.8
Pump, primer, piston, diameter, inches	...	2 1/16	...
circulating, piston, diameter, ins.	...	4	...
stroke, inches	...	14	...
Connecting rod, length between centers, inches	35	35	35
Area of cooling surface for discharge from compressor, square feet	...	50	...
Pipe, diameter, inches :			
Steam	3	...	...
Exhaust	4	...	...
Return air	...	3	...
Discharge air	...	2⅝	...
Receive air	...	...	2½
Discharge air	...	...	3

TEST OF REFRIGERATING PLANT.

At the beginning of this test the ice machine was in operation, but the refrigerating air distributing valves were not open. Observations were made every thirty minutes. In the refrigerator rooms the change in temperature only was noted; for the scuttle butts, temperature of water admitted and drawn off and the number of gallons drawn off; for the ice tank, the temperature of the water initially and the time elapsed to completely freeze this water. The ice tank has provision for nine ice-cake cans. The refrigerating air was sent through the ice tank, refrigerator rooms and scuttle butts successively. The primer pump maintained an excess of air at all times. All pipes were heavily lagged, as well as those in the refrigerator rooms to the distributing valves. Duration of test, four hours.

DATA OF TEST.

Mean pressure, steam, gauge, pounds	124
compression, gauge, pounds	243
expansion, gauge, pounds	67
number of expansions for air	3.15
revolutions per minute	102
temperature of ice-machine room, degrees Fahrenheit	70
atmosphere, degrees Fahrenheit	43
injection water, degrees Fahrenheit	38
discharge water, degrees Fahrenheit	44
indicated horsepower, estimated	41
Mean piston displacement, in cubic feet, per minute :	
Circulating pump	10
Steam cylinder	128
Compression cylinder	102
Expansion cylinder	70
Primer pump	5
Mean velocity through nozzles, in feet, per minute :	
Suction	200
Discharge	460
Steam	2,600
Exhaust	1,460
Compressor suction	2,000
discharge	3,200
Expander suction	2,200
discharge	1,420
Primer suction	5,400
discharge	1,520

Scuttle butt :		
Mean temperature of water, initial, degrees Fahrenheit		48
drawn off, degrees Fahrenheit		41
gallons water drawn off per hour		98
rate gallons of water drawn off per 24 hours		2,300
Ice tank :		
Temperature of water to be frozen, initial, degrees Fahrenheit		58
Time required to freeze the nine cakes, hours		2.67
Weight of nine cakes of ice, pounds		130
Rate, in pounds, of ice, per 24 hours		1,200
Refrigerator rooms :	*Initial.*	*Final.*
Temperatures, in degrees Fahrenheit, crew	59	19
ward room	55	23
warrant officers	50	26
junior officers	49	20
Captain	49	21
Admiral	55	24

ELECTRICAL EQUIPMENT.

The electrical plant is in the forward end of the engine compartment below the protective deck. It is divided by the central ammunition passage, into two watertight rooms. Forward in the bunker space is a room, which communicates with the dynamo room, for the distribution boards. In each dynamo room is the generator-control board, and the distribution boards can be energized from either dynamo room by transverse switches on these boards. All feeders for power, light and searchlights are energized from the distribution boards, and motor generators energize the gun-elevating and turet-turning motors. The turret feeders are in duplicate. There are two motors for turning each turret and one for elevating each gun.

The searchlights are placed one in each upper fighting top, two on the forward bridge and two on a platform on the main mast. Each is a 30-inch 21,000-candlepower light.

The boat crane, winch, turret turning, and gun elevating motors are operated by a controller at the motor and protected by a circuit breaker. The other motors throughout the ship have a starting panel with a rheostat, circuit breaker and a no-voltage release, to protect them.

The long-arm system of watertight doors and hatches is electrically operated either from the bridge or at the door itself.

The ship is lighted by incandescent lamps, for which there are 1,325 fixtures. And in addition there is one arc lamp in each fireroom and two in each engine room of about 1,200 candlepower each. There are also six portable ventilating sets, and forty-eight fans, for officers' quarters.

The following means of communication are installed:

Electric engine telegraph.
Helm order telegraph.
Rudder indicator.
Revolution indicator for main engines.
Thermostat for fire alarm system.
General alarm gongs.
Watertight door and hatch-closing alarm.
Telephone system.
Voice pipes.
Quarters and officers' calls.
Electrical log to pilot house.
Electrical device to operate steam whistle for signals.
Battery order control system.
Truck-light signal system.
Electric-light signal system.
Wireless telegraph.

The systems and appliances are watertight.

DYNAMO ROOMS.

In each dynamo room there are three 100-kilowatt General Electric marine generating sets, and a dynamo condenser mounted on a combined air and circulating pump.

DATA FOR GENERATING SET.

Number per ship	6
Cylinder, H.P., diameter, inches	10
L.P., diameter, inches	18
Stroke, inches	10
Piston rod, diameter, inches	2¼

Crosshead area, square inches	55
pin, diameter, inches	2¾
length, inches	4¼
Connecting rod, length between centers, inches	25
(rectangular section,) inches	1½ by 2¾
Crank shaft, diameter, inches	5
pin, diameter, inches	5
length, inches	6
Valve travel, inches	3¼
H.P., type	Piston.
L.P., type	Balanced flat.

PIPING.

The steam is supplied by an extension from the auxiliary line in the after fireroom. This system is in duplicate on the port and starboard sides. The steam passes through a separator and a reducing valve to the distributing manifold. There are pipes from this manifold to each dynamo engine. The steam and exhaust of the two dynamo rooms are cross connected. There are extensions to each engine from the exhaust line, which leads to the dynamo condensers, and there are extensions from this line to the port and starboard auxiliary exhaust line in the firerooms. The separators drain through "float" traps to the auxiliary exhaust. The cylinders drain to the exhaust line and provision is made to drain them into a bucket if so desired. The condenser pump is the horizontal, simplex, piston type of Blake design, fitted with medium rubber valves and piston rods of rolled Tobin bronze.

DATA FOR DYNAMO CONDENSER.

Number per ship	2
Tubes, number	617
B.W.G, number	16
outside diameter, inch	⅝
cooling surface, square feet	707
length between tube sheets, inches	72
Shell, inside diameter, inches	28
thickness, inch	¼
Nozzles, exhaust, diameter, inches	10
air-pump suction, diameter, inches	5
injection, diameter, inches	5
discharge, diameter, inches	5

DATA FOR PUMP.

Cylinders, steam, diameter, inches	10
air, diameter, inches	12
water, diameter, inches	12
Stroke, inches	12
Piston rod, diameter, inches	1¾
Nozzles, steam, diameter, inches	1½
exhaust, diameter, inches	2
water suction, diameter, inches	6
discharge, diameter inches	5
air suction, diameter, inches	5
discharge, diameter, inches	3½
Ratio of combined area through all suction valves to mean area of piston :	
Air	0.487
Water	0.582

DATA FOR MOTORS.

Purpose.	Number.	Electrical horse-power of one.	Kilowatts.
Ammunition :			
Chain hoist	30	3	...
Conveyor	6	4	...
Whip hoist	2	3.5	...
Turrets :			
Ammunition hoist	4	55	...
Rammer	4	3.5	...
Gun elevating	4	5	...
motor generators	4	...	5
Turret turning	4	20	...
motor generators	2	...	25
Boat cranes :			
Forward, hoist	2	30	...
turning	2	20	...
Aft, hoist	2	50	...
turning	2	30	...
Deck winches	6	30	...
Watertight doors	36	1	...
hatches	4	1	...
Work shop	1	15	...
Ventilating system	27	0.5 to 12	...
Laundry	1	6	...
Pumps	5	1 to 3.5	...
Dough mixer	1	1	...
Dish washer	1	0.75	...
Torpedo air compressor	2	70	...
Total	152		

TEST OF ELECTRIC PLANT.

The generating sets were tested by the Inspector of Equipment, U. S. N., and the information as to volts, ampères, and the corresponding indicated horsepower was obtained from that office. From this data a curve of indicated horsepower and ampères was constructed. The dynamo condenser pump was tested according to the method previously described, when one dynamo engine was in operation.

GENERATING SET No. 1, U. S. A. C. *TENNESSEE*.

Per cent. load	0.00	0.25	0.50	0.75	1.00	1.33
Volts	125	125	125	125	125	125
Ampères	0	200	400	600	800	1,072
Kilo-watts	0	25	50	75	100	134
Electrical horsepower	0	33.5	67.0	100	134	180
Steam gauge reading	150	150	150	150	150	150
Exhaust gauge reading, inches	24	24	24	24	24	24
Revolutions per minute	354	353	352	348	347	342
Indicator spring No. :						
H.P.	60	60	80	80	100	100
L.P.	20	20	20	20	40	40
Combined area, sq. inch. :						
H.P. card	0.38	1.53	2.39	3.45	3.16	4.16
L.P. card	0.48	1.80	2.23	3.78	2.58	4.00
Combined M.E.P. :						
H.P.	7.70	31.8	66.8	92.6	110.0	144.0
L.P.	3.25	12.4	15.6	26.1	35.6	55.6
Combined indicat'd horsepower :						
H.P.	5.26	21.7	45.4	62.2	73.7	95.0
L.P.	7.33	27.9	35.0	57.8	78.6	121.0
Total	12.6	49.6	80.4	120.0	152.0	216.0
Ratio E.H.P. to I.H.P.	...	0.67	0.83	0.83	0.88	0.83
Piston speed, feet, per minute	591	588	587	581	579	571
Piston displacement, cubic ft., per minute :						
H.P.	313	312	311	308	307	303
L.P.	1,130	1,030	1,030	1,020	1,010	1,000
Ratio	3.4	...	...	...	...	...
Velocity through pipe, in feet, per minute :						
Steam	4,680					
Exhaust	5,740					

DYNAMO CONDENSER PUMP, U. S. A. C. *TENNESSEE.*

Time	3'35	3'43	4'18	4'30
Double strokes, per minute	55	43	37	23
Steam gauge reading	150	150	150	150
Exhaust gauge reading, inches	20	19	18	16
Indicator spring, number	100	100	40	40
Area of card, square inches :				
No. 1, air end	1.05	0.96	1.72	1.21
water end	0.73	0.50	1.09	0.83
No. 2, air end	1.27	0.73	1.89	1.19
water end	0.72	0.30	0.88	0.81
Stroke, in feet	1.00	0.99	0.95	0.90
Mean M.E.P. :				
Card No. 1	28.5	23.8	19.3	13.2
2	31.8	17.2	16.7	14.4
Mean	30.2	20.5	18.0	13.8
Theoretical :				
Card No. 1	51	32	29	20
2	78	39	29	20
Mean	50	36	29	20
Ratio $\frac{\text{Actual}}{\text{Theoretical}}$	0.60	.....	0.62	0.66
Steam pressure in valve chests, pounds, guage :				
Card No. 1	56	34	25	14
2	55	34	26	13
Mean	56	34	26	14
I.H.P. :				
Card No. 1	7.22	4.67	3.12	1.26
2	7.82	3.37	2.70	1.37
Mean	7.52	4.02	2.91	1.32
Piston displacement, in cubic feet, per minute :				
Steam	58.8	45.0	37.2	21.8
Water	85.6	66.2	54.8	32.2
Air	85.6	66.2	54.8	32.2
Piston speed, feet, per minute	110	85.2	70.3	41.4
Velocity through nozzle, in feet, per minute :				
Water suction	436	337	279	164
discharge	630	486	402	237
Air suction	630	486	402	237
discharge	1,280	993	822	482
Steam	4,780	3,660	3,020	1,770
Exhaust	2,700	2,070	1,710	1,000

The electrical load was nearly constant throughout trials.

DATA FROM STANDARDIZATION TRIAL.

Mean, ampères	392
corresponding indicated horsepower	82

DATA FROM FOUR-HOUR TRIAL.

Mean, ampères	427
indicated horsepower	88
condenser pump, double strokes per minute	31
corresponding indicated horsepower	2.1
vacuum, inches	16

HEATING SYSTEM.

This is divided into two sections. The aft section has its steam-distributing manifold in the port engine room, where it receives steam from the auxiliary line. Steam passes through a reducing valve to the manifold, to which is attached a relief valve and steam gauge. The drains from the several circuits are to the auxiliary exhaust through expansion traps, or the traps may be passed. Steam to officers' pantries and shower baths are an independent system and drain direct to the auxiliary exhaust.

The manifold for the forward section is in the central ammunition passage and receives steam, through a reducing valve, from the auxiliary line. The drains from the circuits are to the auxiliary exhaust, through expansion traps, or the traps may be passed. The firemen's shower baths are on the same circuit as the galley, and the crew's shower baths are on the same circuit as the sick bay. Both bath drains are direct to the auxiliary exhaust.

OFFICIAL TRIALS.

THE STANDARDIZATION TRIAL.

The U. S. A. C. *Tennessee* left her anchorage at Rockland, Me., about 7 A. M., February 11, 1906, and prepared generally for the trial over the measured-mile course. The draught was noted before and after the trial as follows:

Table I.—U. S. A. C. *TENNESSEE.*

Data from Progressive Trial.

Number and direction of run.	Time on course.	Revolutions of main engines.					Speed in knots per hour.	Indicated horsepower.											
		To make 1 knot.		Per minute.				Port engine.					Starboard engine.					Total of both main engines.	Auxiliaries in operation.	Total for all auxiliaries.
		Port.	Starboard.	Port.	Starboard.	Mean.		H.P. cylinder.	M.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.	H.P. cylinder.	M.P. cylinder.	F.L.P. cylinder.	A.L.P. cylinder.	Total.			
	m. s.																			
1–N.	2–44.2	340.3	339.4	124.3	123.9	124.1	21.92	4,174	3,991	2,131	2,092	12,387	4,271	4,243	2,125	2,125	12,763	25,150	986	27,140
2–S.	2–47.4	346.6	345.4	124.2	123.8	124.0	21.51	4,070	3,875	2,091	2,052	12,088	4,175	4,004	1,994	2,058	12,231	24,320	982	25,300
3–N.	2–42.1	340.5	339.9	126.0	125.8	125.9	22.21	4,337	4,124	2,213	2,174	12,848	4,433	4,404	2,157	2,222	13,216	26,060	942	27,010
4–S.	2–44.6	348.4	348.0	127.1	126.9	127.0	21.87	4,338	4,061	2,205	2,179	12,783	4,437	4,345	2,110	2,242	13,134	25,920	989	26,910
5–N.	2–41.0	339.0	341.2	126.4	127.2	126.8	22.36	4,245	4,039	2,232	2,219	12,735	4,149	4,500	2,313	2,299	13,260	26,000	912	26,910
6–S.	2–53.0	334.0	344.5	115.8	119.4	117.6	20.81	3,298	2,871	1,540	1,552	9,261	3,846	3,544	1,639	1,826	10,855	20,120	660	20,780
7–N.	2–53.3	327.9	333.8	113.6	115.7	114.6	20.78	3,097	2,533	1,535	1,523	8,688	3,598	3,230	1,586	1,598	10,013	18,700	545	19,250
8–S.	2–58.5	332.6	334.1	111.5	112.0	111.8	20.17	3,008	2,444	1,344	1,332	8,128	3,223	2,728	1,374	1,432	8,756	16,880	397	17,280
9–N.	3–01.0	326.7	330.9	108.3	109.7	109.0	19.89	2,683	2,274	1,227	1,204	7,388	3,019	2,670	1,288	1,333	8,311	15,700	392	16,090
10–S.	3–03.5	321.9	319.8	105.0	104.4	104.7	19.62	2,545	2,038	1,179	1,168	6,929	2,695	2,025	1,182	1,160	7,061	13,990	475	14,470
11–N.	3–21.6	337.0	336.0	100.4	100.1	100.2	17.85	2,349	1,909	1,022	970	6,250	2,355	1,903	873	936	6,068	12,320	339	12,660
12–S.	3–15.4	315.7	313.0	97.3	96.5	96.9	18.42	1,956	1,651	890	829	5,326	2,058	1,725	812	862	5,457	10,780	334	11,120
13–N.	3–37.0	335.9	335.9	92.9	92.8	92.9	16.59	1,802	1,413	753	724	4,691	1,904	1,568	695	724	4,891	9,582	334	9,916
14–S.	3–28.4	311.5	308.2	89.8	88.8	89.3	17.27	1,594	1,168	616	653	4,030	1,724	1,423	642	660	4,448	8,478	285	8,763

Table II.—U. S. A. C. *TENNESSEE.*

Data from Progressive Trial.

Number and direction of run.	Which engine.	Steam pressure.			Mean effective pressure.				Vacuum, in inches.	Air pressure, in inches of water.	Cut-off of cylinders from beginning.			Temperatures.				Revolutions or double strokes.								
		In boilers above atmosphere.	At engine above atmosphere.	H.P. steam chest, absolute.	In 1st receiver, absolute.	In 2d receiver, absolute.	H.P. cylinder.	I.P. cylinder.	Forward L.P. cylinder.	After L.P. cylinder.	Referred to L.P.			H.P.	I.P.	Forward L.P.	After L.P.	Engine-room working platform.	Fireroom working level.	Injection.	Discharge.	Hotwell.	Feed.	Generating set.	Circulating pump.	Main feed pump.	Feed supply pump.	Sanitary pump.	Bilge pump.	Air pump.	Dynamo condenser pump.	Fireroom blower.
1—N.	S.	271	260	253	108	39	125	45	16.5	16.5	49.57	26	26	.78	.76	.73	.73	50	44	34	110	64	220	350	156	28	38	54	21	19	600
1—S.	P.	271	257	245	100	38	122	42	16.5	16.2	47.96	26	26	.78	.76	.73	.73	68	44	36	108	64	184	0	150	27	11	38	22	0	525
2—S.	S.	264	247	240	108	37	123	43	15.5	16.0	47.54	26	1.9	.78	.76	.73	.73	50	47	34	110	64	216	350	146	23	25	34	21	19	609
2—S.	P.	259	240	240	37	39	119	41	16.2	15.9	46.82	26		.78	.76	.73	.73	70	47	38	107	64	170	0	160	27	24	36	21	19	543
3—N.	S.	277	265	260	110	40	138	46	16.5	17.0	50.55	26	1.9	.78	.76	.73	.73	56	48	34	110	64	234	350	152	25	24	34	21	19	610
3—N.	P.	277	266	255	103	40	135	43	16.9	16.6	49.04	26		.78	.76	.73	.73	70	48	36	108	60	190	0	160	29	11	42	23	19	536
4—S.	S.	275	264	257	110	41	137	45	16.0	16.5	49.80	26	1.9	.78	.76	.73	.73	58	48	34	112	64	232	350	150	23	30	54	27	19	614
4—S.	P.	273	265	255	105	40	134	43	16.7	16.5	48.40	26		.78	.76	.73	.73	70	48	36	108	60	190	0	158	25	20	40	24	19	548
5—N.	S.	276	265	266	115	42	138	48	17.5	17.4	50.18	26	2.0	.78	.76	.73	.73	50	47	34	112	64	220	350	152	28.5	28	54	22	19	504
5—N.	P.	276	265	255	104	41	138	48	17.0	16.9	48.48	27		.78	.76	.73	.73	70	47	36	105	62	198	57	195	27	28	42	22	0	558
6—S.	S.	259	260	230	94	33	117	39	13.8	14.7	43.73	26	0.7	.78	.76	.73	.73	54	47	34	118	62	200	350	170	22.5	24	42	21	19	368
6—S.	P.	259	260	205	75	30	104	33	12.8	12.9	38.49	27		.78	.76	.73	.73	72	47	36	90	60	170	0	168	24	22	43	26	0	350
7—N.	S.	261	255	222	90	33	113	37	13.2	13.3	41.67	26	0.3	.78	.76	.73	.73	60	47	35	104	54	198	350	146	20.5	22	42	22	19	398
7—N.	P.	261	259	200	75	30	99	29	13.0	12.9	36.79	26		.78	.76	.73	.73	74	47	36	96	60	180	0	144	21	22	48	22	0	248
8—S.	S.	247	249	200	77	28	105	32	11.8	12.3	37.01	27	0.1	.78	.76	.73	.73	64	47	34	105	53	208	350	130	16	28	42	21	19	270
8—S.	P.	247	245	190	70	28	100	29	11.6	11.5	35.07	27		.78	.76	.73	.73	72	47	36	95	56	150	0	130	22	18	52	18	0	244
9—N.	S.	243	239	195	75	27	100	32	11.3	11.7	36.46	27	0.1	.78	.76	.73	.73	56	50	35	104	56	210	350	126	14	21	52	24	19	239
9—N.	P.	243	244	175	65	25	90	28	10.9	10.7	32.82	27		.78	.76	.73	.73	74	50	36	108	58	170	0	110	19	16	48	24	0	237
10—S.	S.	269	263	178	66	25	94	26	10.9	10.7	32.57	27	0.1	.78	.76	.73	.73	68	51	34	100	49	210	350	130	24	19	46	26	19	207
10—S.	P.	269	270	170	62	25	88	26	10.8	10.7	31.75	27		.78	.76	.73	.73	76	51	36	98	65	215	0	158	24	18	58	30	19	276
11—N.	S.	256	254	157	57	21	86	25	8.4	9.0	29.18	27	0.0	.78	.76	.73	.73	72	50	35	103	50	210	350	100	13	9	58	23	19	175
11—N.	P.	256	252	155	55	22	85	25	9.8	9.3	29.96	26		.78	.76	.73	.73	80	50	36	92	64	188	0	108	15	14	58	23	19	254
12—S.	S.	248	245	145	53	20	75	24	8.1	8.6	27.21	27	0.0	.78	.76	.73	.73	69	52	35	104	50	196	350	100	15	15	58	26	19	156
12—S.	P.	248	250	143	52	20	73	22	8.8	8.2	26.33	26		.78	.76	.73	.73	78	52	36	89	61	168	0	105	15	10	60	15	19	238
13—N.	S.	259	245	138	49	18	75	22	7.2	7.3	25.35	27	0.0	.78	.76	.73	.73	68	52	34	103	50	194	350	106	9.5	11	56	25	19	080
13—N.	P.	259	260	130	45	19	71	20	7.8	7.5	24.31	27		.78	.76	.73	.73	80	52	36	90	62	158	0	100	9.5	10	60	34	19	232
14—S.	S.	254	245	102	43	18	71	21	7.0	7.2	24.26	27	0.0	.78	.76	.73	.73	70	52	34	104	50	194	350	48	9.5	12	57	18	19	080
14—S.	P.	254	255	115	41	18	65	17	6.6	7.0	21.60	27		.78	.76	.73	.73	79	52	36	81	60	174	0	150	11	13	60	18	0	000

	Before.	After.
Forward, feet	24.83	24.39
Aft, feet	25.30	25.29
Mean, feet	25.07	24.84
Corresponding displacement, in tons	14,580	14,380

The first run began at 8h—45m—53.6s A. M. and the last one at 12h—18m—3.6s M. In all fourteen runs over the course were made, five at maximum and the others at successively lower speeds. A curve showing the relation between the revolutions per minute of the propellers and the speed of the ship in knots was plotted from this data, and the revolutions necessary to maintain a speed of 22 knots was determined to be 126.2.

The curves illustrated here were obtained as follows: The R.p.m.-knots from the Board of Inspection and Survey, R.p.m.-I.h.p. from the Bureau of Steam Engineering and the I.h.p.-knots is a combination of the two.

During the trial the machinery operated smoothly and there was very little vibration. The performance of the auxiliaries has been combined with their description, and of the firerooms is given under the heading "Flue Gas Analysis."

After the trial the *Tennessee* returned to Rockland, when several tests not under the cognizance of the Bureau of Steam Engineering were made.

THE FOUR-HOUR TRIAL.

This trial was made on February 12, while the ship was under way for Boston. The weather was generally favorable. On this trial the difficulty previously experienced in getting sufficient steam was overcome, and at no time was there a deficiency. The machinery operated smoothly throughout the trial. The only accident was the breaking of the head of the starboard feed-water heater. The mean draught during the trial was 25 feet and the corresponding displacement 14,500 tons.

DATA OF STEAM GAUGE READINGS.

Observation number.	Fireroom.		At Engine.		H. P. Receiver.	
	P.	*S.*	*P.*	*S.*	*P.*	*S.*
1	.....	.....	265	.....	248	.. ..
2	278	278	262	265	248	249
3	278	278	265	265	253	251
4	282	282	265	268	249	252
5	276	276	265	265	253	252
6	279	279	272	260	253	252
7	273	273	262	260	252	247
8	274	274	272	265	248	247
9	278	278	270	265	253	251
Mean	277	277	266	264	250	250

	M. P. Receiver.		L. P. Receiver.		Vacuum.	
	P.	*S.*	*P.*	*S.*	*P.*	*S.*
1	87	...	24	...	26	...
2	88	105	25	28	26	26
3	92	99	26	26	26	26
4	90	95	27	34	26	26
5	90	105	29	30	26	26
6	92	103	29	29	26	26
7	88	98	27	28	26	26
8	92	98	29	28	26	26
9	92	101	28	29	26	26
Mean	90	100	27	27	26	26

DATA OF MAIN ENGINES.

INDICATED HORSEPOWER.

Observation number.	*Port Engine.*				
	H.P.	*L.P.*	*F.L.P.*	*A.L.P.*	*Total.*
1	4,366	4,200	2,255	2,320	13,141
2	4,408	3,597	2,113	2,258	12,376
3	4,475	4,171	2,146	2,212	13,004
4	4,352	3,977	2,292	2,384	13,005
5	4,334	3,944	2,327	2,381	12,986
6	4,202	4,065	2,221	2,247	12,735
7	4,333	4,111	2,259	2,352	13,055
8	4,420	4,167	2,313	2,366	13,236
Mean	4,361	4,029	2,241	2,315	12,950

Total for L.P., 4,556.

Observation number.	*Starboard Engine.*				
	H.P.	*L.P.*	*F.L.P.*	*A.L.P.*	*Total.*
1	4,109	4,929	2,376	2,402	13,816
2	4,203	4,550	2,247	2,247	13,247
3	4,492	4,348	2,217	2,177	13,234
4	4,241	4,670	2,511	2,445	13,867

Observation number.	*Starboard Engine.*				
	H.P.	*L.P.*	*F L.P.*	*A.L.P.*	*Total.*
5	4,205	4,786	2,534	2,468	13,993
6	4,317	4,414	2,385	2,385	13,501
7	4,225	4,409	2,382	2,369	13,385
8	4,356	4,417	2,453	2,440	13,666
Mean	4,269	4,565	2,388	2,367	13,590

Total for L.P., 4,755.

Observation number.	*Revolutions per Minute.*			
	P.	*S.*	*Mean.*	*Knots.*
1	126.2	126.2	126.2	22.01
2	127.1	126.9	127.0	22.12
3	127.3	127.5	127.4	22.16
4	127.4	127.2	127.3	22.15
5	128.3	127.7	128.0	22.24
6	128.2	128.0	128.1	22.25
7	126.6	126.6	126.6	22.08
8	127.6	128.0	127.8	22.23
Mean	127.4	127.3	127.4	22.16

The pyrometer readings were taken from stacks numbered 1, 2 and 3, but No. 4, the after stack, was omitted. Two pyrometers, one of the contractors' and one of the Trial Board's, were used in the following manner. Each pyrometer was inserted in the same stack and the variation found to be 100 degrees Fahrenheit.

After the reading of stack No. 1 the pyrometer was inserted in stack No. 3, and so on. By this method the pyrometer had been in the stack for one-half hour previous to reading. Since the trial trip the pyrometer of the Bureau of Steam Enginering has been calibrated, and the readings recorded below are the corrected readings:

Pyrometer reading degrees Fahrenheit.	Stack No. 1.	Stack No. 2.	Stack No. 3.	
	603	623	593	
	638	615	613	
	623	613	616	
	638	...	620	
	648	...	618	
	643	...	608	
	643	...	598	
	...	...	617	
Mean	633	617	610	
of three stacks				620

TEMPERATURES, DEGREES FAHRENHEIT.

Mean Temperatures.	*Port.*	*Starboard.*
Injection	42	42
Discharge	109	112
Feed tank	70	63
Feed water	198	121
Engine rooms, upper platform	87	74
working platform	72	69
Fire rooms, at gratings	109	109
floors	74	74

AUXILIARIES.

	Mean revolutions, or double strokes, per minute.		Mean indicated horsepower.	
	Port.	*Starboard.*	*Port.*	*Starboard.*
Main circulating pumps	173	174	71	72
air pumps	20	21	21	21
feed pumps	27	27	134	133
Feed-supply pumps	25	25	10	9
Fire and bilge pumps	44	44	6	5
Forced-draft blowers	484	488	157	162
Auxiliary feed pumps	0	0	0	0
Generating set	.....	.....	88	0
Dynamo condenser pump	31	0	2	0
Auxiliary condenser pumps	0	0	0	0

SUMMARY OF INDICATED HORSEPOWER.

Main engines, port	12,950
starboard	13,590
total	26,540
difference	640

Totals for Port and Starboard.

Main circulating pumps	143
air pumps	42
feed pumps	267
Feed supply pumps	19
Fire and bilge pumps	11
Forced-draft blowers	319
Auxiliary feed pumps	0
Generating set	88
Dynamo condenser pump	2
Auxiliary condenser	0
Total for auxilaries	891
Total for main and auxiliary engines	27,430

COAL.

Kind and quality.....Pocahontas run of the mine, fair.
Pounds per hour.....Not determined.

DEDUCED DATA.

I.h.p. per square foot of grate surface :	
Main engines	16.6
and auxiliaries	17.1
I.h.p. per square foot of heating surface :	
Main engines	0.374
and auxiliaries	0.387
Square feet of heating surface per :	
I.h.p. of main engines	2.67
and auxilaries	2.58
Square feet of cooling surface per :	
I.h.p. of main engines	1.11
and auxilaries	1.07
Ratio of heating to cooling surface	2.42

The indicated horsepower of the auxiliaries was determined from the curves obtained from the tests. In preparing these curves the writer wishes to acknowledge the assistance of Mr. G. M. Spear, S. B., of the Wm. Cramp & Sons S. & E. B. Co.

FLUE GAS ANALYSIS.

The subject of fireroom management for official trials has received a great deal of attention, especially since the introduction of water-tube boilers, which require radically different methods from the Scotch type of boiler. Various air pressures and thicknesses of fires have been tried until it is generally accepted that the fires should be about 6 to 8 inches thick, maintained at a uniform thickness, fired in small amounts and at frequent intervals, with an air pressure of 1.5 to 2 inches of water. As the writer could find no data of flue-gas analyses taken on trial trips, arrangements were made for this purpose. It was the intention to take samples of the coal for a calorific determination and an ultimate analysis if it could be arranged for; but this was abandoned, because the weight of coal burned on the four-hour trial was not obtained and the coal was a mixture of Pocahontas run-of-the-mine of fair quality,

and New River in somewhat variable proportions. The coal consumption of ships has been compared by the pounds of coal burned although no calorific determination has been made, and it was thought that perhaps the introduction of this information would be interesting for making more exact comparisons.

It has been the custom of the Bureau of Steam Engineering to have photographs taken at regular intervals, of the smoke emitted from the stacks. This, of course, is only an observation of condition, and in the case of this trial trip clearly shows that a large excess of air reduces the density of the smoke. The four-hour trial smoke is denser than on the standardization trial, which may be due, in part, to the fact that there was present in the smoke 55 per cent. more air than on the four-hour trial, and its function may be largely dilution. It would require a careful test to determine the percentage of carbon present in the smoke in each case. If there is more unconsumed carbon in the denser smoke it would be interesting to determine the heat lost by the unconsumed carbon compared with the heat lost by the excess air in the lighter smoke.

The gas analysis of the two cases indicated that the lower per cent. of CO_2 gave the lighter smoke, and it seems that as the heat supplied the boilers varies directly with the percentage CO_2 in the waste gases the determination of the CO_2 would control the system of firing perfectly, *i. e.*, the firing interval, amount of coal fired, air pressure and thickness of fires. It is, of course, necessary to maintain the fires at a uniform thickness to prevent the air flowing through the place of least resistance and producing an excess of air. As the boiler is expected to evaporate water at a uniform rate, a uniform supply of heat—indicated by a uniform per cent. of CO_2—would represent the most desirable condition.

The gas analyses made on these trials were by an Orsat apparatus, which requires about twenty minutes for the complete analysis but only a few minutes to determine the CO_2. There are several devices to determine the per

cent. of CO_2 continuously. One of them depends on the difference of the specific gravity of air and CO_2. The smoke is drawn through a reservoir in which is a hollow sphere filled with air and attached to a lever. It is free to rotate, and by this means moves a pointer. Another device is on the principle of the pneumatic pyrometer which measures temperatures by the contraction of heated air, but this depends on the absorption of the CO_2 instead of the air contraction.

For the purpose of making the gas analyses the loan of an Orsat apparatus from the Wm. Cramp & Sons S. & E. B. Co. was obtained by Commander A. V. Zane, U. S. N. A second one was loaned from the laboratories of the Massachusetts Institute of Technology, through the courtesy of Mr. C. F. Willard and Dr. A. H. Gill of the Faculty. The return of this apparatus to the Institute was very kindly arranged for by Captain Wm. S. Moore, U. S. N. The writer also wishes to acknowledge the assistance of Mr. E. H. Peabody, Engineer in Charge of the Department of Tests for the Babcock & Wilcox Co., Mr. C. R. Spare, Chemist for the Wm. Cramp & Sons S. & E. B. Co., and Mr. H. E. Cade, of the Office of the Inspector of Equipment, U. S. N. Mr. Cade operated one Orsat apparatus.

The necessary data for complete computations could not be obtained; hence the characteristics of one pound of carbon have been taken, and no attempt was made to assume the chemical and calorific determinations of the coal used. As there was no pyrometer in the after stack, it is not possible to determine the relative stack temperatures for the comparison of boilers *D* and *N*, but it was assumed to be the same as the other three stacks.

DESCRIPTION OF APPARATUS.

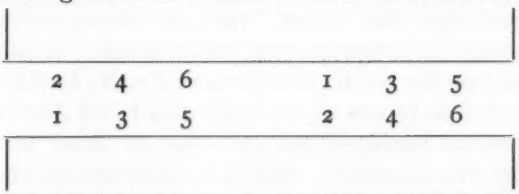
"The apparatus was the usual Orsat type, and the reagents used were as follows:

Reagent.	To absorb.	Absorption per 1 cc.
Potassium hydrate	CO_2	40 cc. of CO_2
pyrogallate	O	2 cc. of O.
Acid cuprous chloride	CO	1 cc. of CO.

The apparatus was fastened to the outboard side of the boiler, and a ¼-inch pipe, unperforated, about 3½ feet long, inserted in the top tube-cleaning hole of the boiler casing. This is very near the point where the flue gases leave the boiler. The volume of gas contained in the pipe was determined and considered when taking a sample of gas. The gas was drawn directly into the Orsat apparatus and immediately analyzed.

After some experimenting for general information the samples of gas were taken just before firing the middle furnace.

The arrangement of furnace doors is sketched below:



As the firing interval was one minute, each furnace was fired once in 6 minutes. At the time the sample was taken the outboard furnace had been fired 4 minutes, the middle one almost 6, and the next one 2 minutes. This would represent the condition of best combustion as near as we could determine.

The method used for computing the weight of air used per pound of carbon is based on the atomic weights of carbon and oxygen, and the chemical combination of these two elements are represented by the two equations:

(1) $C + 2O = CO_2$

Atomic weight $12 + 32 = 44$,

Hence, a given weight of CO_2 will contain

$\frac{12}{44}$ of C and $\frac{32}{44}$ of O.

(2) $C + O = CO$

Atomic weight $12 + 16 = 28$.

Hence, a given weight of CO will contain

$\frac{12}{28}$ of C and $\frac{16}{28}$ of O.

The volume resulting from the combining of oxygen and carbon is equal to the original volume of oxygen. Hence the absorption by the Orsat apparatus should be approximately equal to the amount of oxygen present in the air at the place where the analysis is made. Part of the oxygen will combine with the other elements in the coal and the Orsat absorption will be reduced by this amount. · Oxygen is present in the air in the following proportions:

By volume, 21 per cent.
weight, 23 per cent.

The weight for 1 cubic centimeter
of CO_2 gas = 0.01966 grams,
O gas = 0.00143 grams,
CO gas = 0.00125 grams.

The computation would be as follows: the volume of each gas is given in cubic centimeters and the computed weights of the elements will be in grams.

CO_2

$$\text{Weight of O} = 0.01966 \times \frac{32}{44} CO_2 = 0.001430\ CO_2$$

$$C = 0.01966 \times \frac{12}{44} CO_2 = 0.000536\ CO_2.$$

O

$$\text{Weight of O} = 0.00143 \times O.$$

CO

$$\text{Weight of O} = 0.00125 \times \frac{16}{28} CO = 0.000714\ CO.$$

$$C = 0.00125 \times \frac{12}{28} CO = 0.000536\ CO.$$

From the above the total weight of O and C may be obtained.

Since air contains about 23 per cent. of O by weight the number of grams of air required to supply the weight of O as determined above would be:

$$\text{Grams of air required} = \frac{\text{grams of O}}{0.23} = \text{A}.$$

$$\text{And} \left\{ \begin{matrix} \text{pounds} \\ \text{grams} \end{matrix} \right\} \text{of air per} \left\{ \begin{matrix} \text{pounds} \\ \text{grams} \end{matrix} \right\} \text{of carbon} = \frac{\text{A}}{\text{Grams of C, (as determined above)}}.$$

The heat rejected to stack by excess air is computed by determining, as above, the air used per pound of carbon. The difference between this and 11.6 pounds—the weight of air required theoretically to turn one pound of carbon to CO_2—will be the excess air present in the flue gases. The specific heat of air is taken as 0.24. The fireroom and stack temperatures were observed.

STANDARDIZATION TRIAL.

The conditions generally were those which usually obtain for this type of boiler. The fires were 6 to 8 inches thick, air pressure equivalent to about 2 inches of water and the firing interval about 1 minute. This produced sufficient steam to complete the trial, but almost no reserve. The firemen were worked hard, and the blower engines made about 600 revolutions per minute. The Orsat analysis during this trial are given below. The analyses were not continued after the maximum speed test had been completed, except a few analyses, which clearly indicated that with less air pressure the combustion was greatly improved.

Observation number.	1	2	3	4	5	6
Per cent. CO_2	8.4	8.2	8.0	8.7	11.6	11.8
O	12.4	12.0	11.5	11.1	6.4	6.3
CO	0.4	0.2	0.0	0.1	0.0	0.3
Total	21.2	20.4	19.5	19.9	18.0	18.4

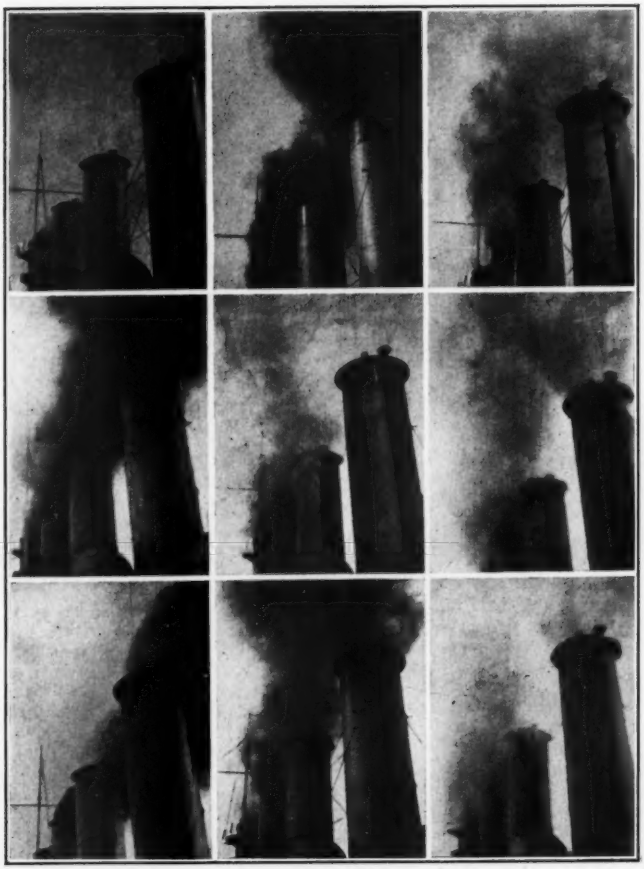
Notes :—

No. 1 and 2 taken just after firing middle furnace.

No. 3 and 4 taken just before firing middle furnace.

No. 5 and 6 taken just before firing middle furnace and under natural draft with fires about 8 inches thick.

Firing interval about one minute.



PHOTOGRAPHS OF SMOKE TAKEN DURING TRIALS, U. S. S. "TENNESSEE."

Conclusion.—Fires too thin and perhaps burned through in places, admitting a large excess of air, thus reducing the effective heat of the furnace. Thicker fires should give better results.

FOUR-HOUR TRIAL.

Conditions in the firerooms were changed for this trial by reducing the air pressure from 2 to 1.5 inches of water and increasing the thickness of the fires from 6 or 8 inches to 12 or 14 inches, while the firing interval remained one minute.

Previous to putting coal on the fires the firemen raked them to nearly a uniform thickness, but probably thickest at the bridge wall. The coal was spread uniformly over the grates to about .75 of the way back. By this arrangement the volatile gases driven off at first pass to the back of the furnace and come in contact with the thoroughly incandescent coal, which has been raked back, and the hot bridge-wall, thus completing the combustion. The smoke photographs taken indicate that the smoke was more dense than on the previous trial. It was the writer's desire that some of the smoke photographs be taken from the quarter deck, in order that the smoke might be compared with parts of the ship, and perhaps a better idea of the smoke obtained. Photograph number one was taken from this position, but the others from the usual position.

With the changed conditions in the firerooms there was no difficulty in supplying all the steam required by the main engines. The safety valves lifted several times and the main engines were making at one time 130 revolutions per minute with steam to receivers, and this condition could probably have been maintained for the entire trial had it been permitted. After 1½ hours a head of the starboard feed-water heater broke, and the temperature of the feed water was reduced from about 220 degrees Fahrenheit to about 65 degrees Fahrenheit, but in the fireroom the revolutions of the blowers were being continually reduced

with a corresponding reduction in air pressure. This fireroom condition easily maintained the revolutions of the main engines at 127.4.

An interesting comparison was noted in the firerooms for boilers *D* and *N*. Each fireman tended the furnace door for his mate on boiler *N* but on boiler *D* each fireman tended the door for himself. In the latter case the furnace doors were open longer than the other. The system for boiler *D* was used for most of the boilers. The Orsat analyses are given below. The per cent. of CO was so small that it could not be read on the burette on account of the vibrations of the ship, and if there were any it was practically negligible, because the total absorption indicates that this was very nearly complete.

The computations are recorded by two and in some cases three significant figures, which allows for a possible error of 10 per cent., as stated at the beginning of the article. The stack temperature for the standardization trial is the mean of one observation only, of two stacks, and only one significant is used to record it.

	Boiler D.			*Boiler N.*		
Observation Number.	CO_2	O	Total.	CO_2	O	Total.
1	12.0	8.2	20.2	11.9	6.0	17.9
2	9.0	11.2	20.2	13.4	6.0	19.4
3	12.0	7.4	19.4	15.3	3.9	19.2
4	11.2	8.8	20.0	14.6	5.3	19.9
5	13.6	5.2	18.8	15.1	4.5	19.6
6	12.6	5.6	18.2	15.1	3.4	18.5
7	11.2	9.0	20.2	13.0	6.2	19.2
Mean	11.7	7.9	19.6	14.1	5.2	19.3

COMPARISON DATA.

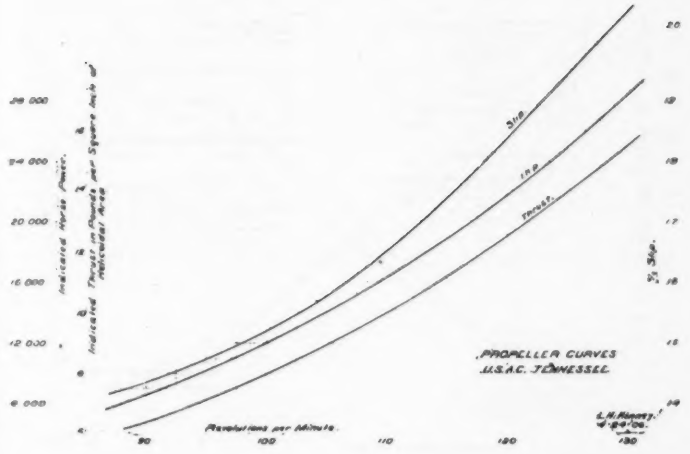
Means:	*Boiler D.*	*Boiler N.*
Firing interval, minutes	1	1
Furnace doors open, seconds	50	40
Thickness of fires, inches	12 to 14	
Air pressure, inches of water	1.47	
pounds of, per pound of C	19.5	15.9
excess	7.9	4.3
times excess over theoretical	1.68	1.37
Air per cent. less	...	18

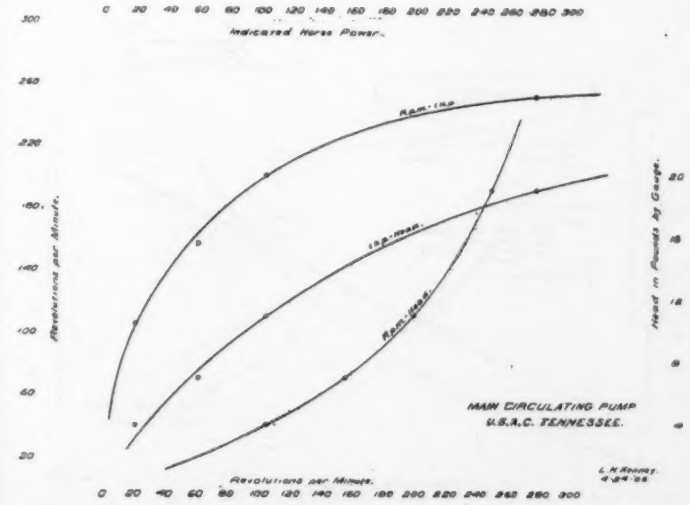
Means :		
Air per cent. more	22	...
B.t.u. rejected to stack, by excess air, per pound of C. (stack temperature assumed the same).	1,000	560
per cent. less	...	44
more	79	...
of the heat theoretically available from one lb. of C (14,500 B.t.u. assumed)	6.9	3.9

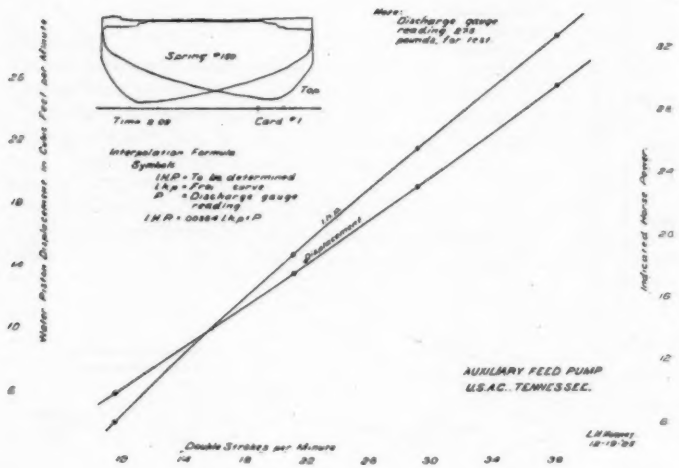
COMPARISON OF TRIALS.

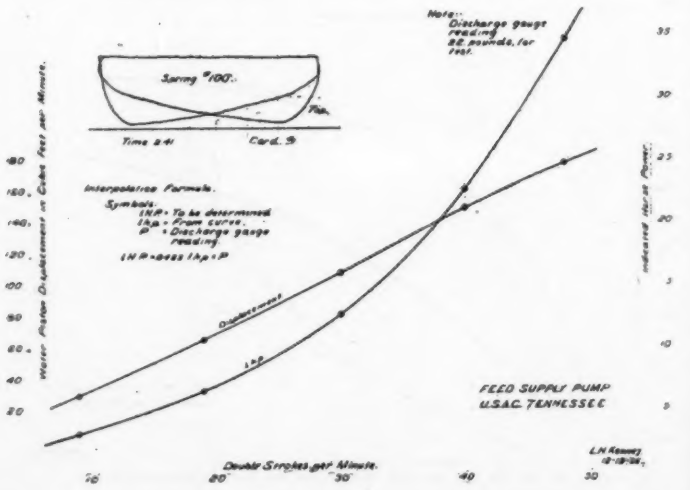
Means :	Standardization, maximum runs only.	Four-hour.
Firing interval, minutes	1	1
Furnace doors open, seconds	50	50
Thickness of fires, inches	6 to 8	12 to 14
Air pressure, inches of water	1.94	1.47
per cent. less	...	24
more	32	...
Blowers :		
Revolutions per minute	574	486
Indicated horsepower	438	304
per cent. less	...	30
more	44	...
Steam consumption per hour, pounds, (60 lbs. per i.h.p.-hr. assumed)	25,000	18,000
Pounds less	..	7,000
Gas analyses :		
CO_2	8.3	12.9
O	11.8	6.6
CO	0.2	0.0
Air :		
Pounds of, per pound of C	27.5	17.7
excess	15.9	6.1
times excess over theoretical	2.37	1.53
per cent. less	...	35
more	55	...
Temperature, firerooms, degrees F	50	74
stack (mean of Nos. 1, 2 and 3.)	600	620
B.t.u. in excess air :		
Rejected to stack per pound of C	2,100	800
Per cent. less	...	24
more	62	...
of the heat theoretically available from one pound of C, (14,500 B.t.u. assumed)	14	5.5
per cent. loss due to CO	1	0

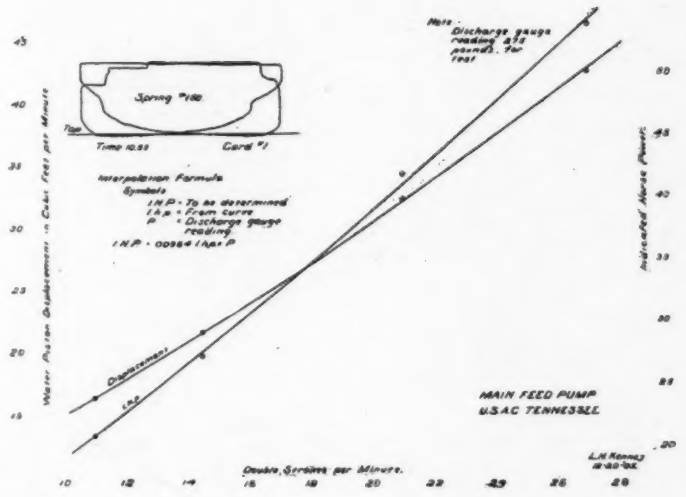
	Standardization, maximum runs only.	Four-hour.
Maximum:		
Steam pressure in boilers, gauge	276	283
Revolutions per minute	127	130
Indicated horsepower	26,000	28,300
Speed in knots	22.10	22.44
per cent. more	...	1.5
Mean of runs 1 to 5:		
Revolutions	125.6	127.4
Indicated horsepower	25,500	26,500
Speed	21.93	22.16

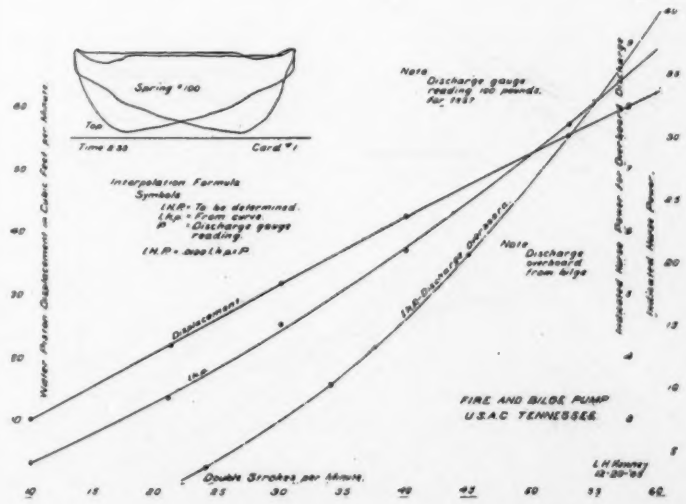

Indicated Horse Power.
Indicated Thrust in Pounds per Square Inch of Indicated Area
Revolutions per Minute.
Slip.
PROPELLER CURVES
U.S.A.C. TENNESSEE.
L. M. Rowell
4-20-05.

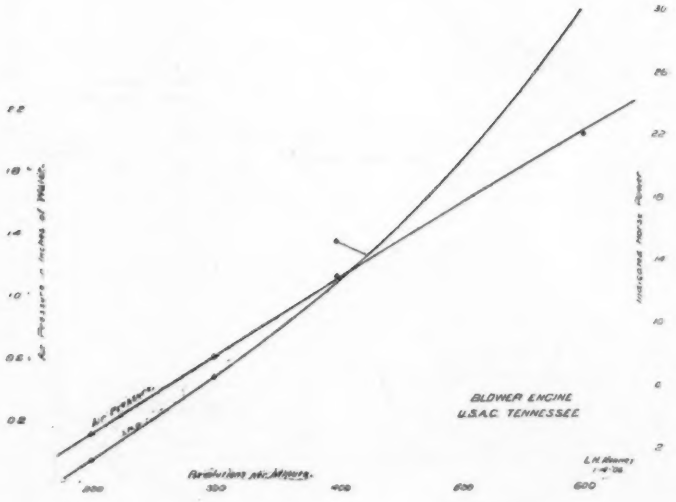

Indicated Horse Power.
Revolutions per Minute.
Head in Pounds by Gauge.
MAIN CIRCULATING PUMP
U.S.A.C. TENNESSEE.
L. M. Rowell
4-24-05

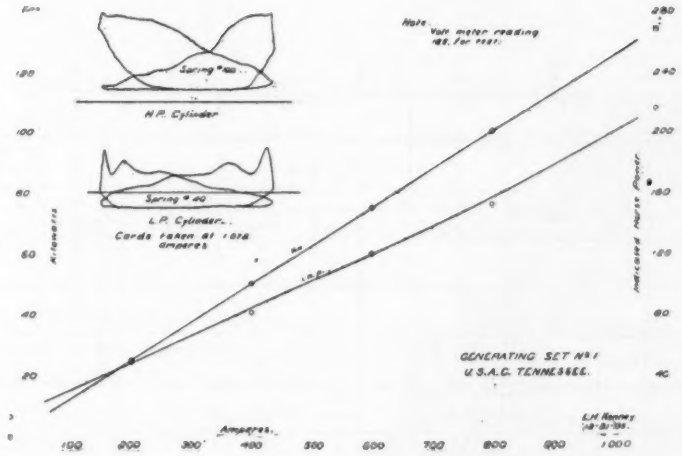

Note:
Discharge gauge reading 875 pounds, for test.
Spring #150
Top
Time 2.09
Card #1
Interpolation Formula
Symbols
I.H.P. = To be determined
I.h.p. = From curve
P = Discharge gauge reading
I.H.P. = constant I.h.p. + P
I.H.P.
Displacement
AUXILIARY FEED PUMP
U.S.A.C. TENNESSEE.
Water Piston Displacement in Cubic Feet per Minute
Indicated Horse Power
Double Strokes per Minute
L.H. Kenney
12-19-04

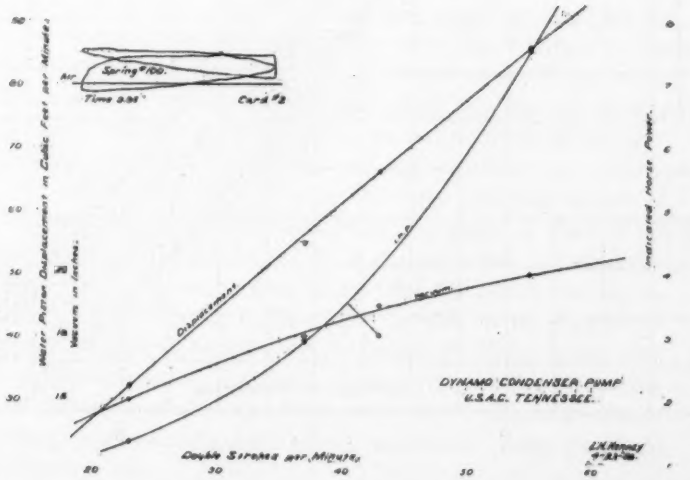

Note:
Discharge gauge reading 82 pounds, for test.
Spring #100
Top
Time 2.41
Card. 9
Interpolation Formula.
Symbols
I.H.P. = To be determined.
I.h.p. = From curve.
P = Discharge gauge reading.
I.H.P. = const. I.h.p. + P
Displacement
I.H.P.
FEED SUPPLY PUMP
U.S.A.C. TENNESSEE
Water Piston Displacement in Cubic Feet per Minute.
Indicated Horse Power.
Double Strokes per Minute.
L.H. Kenney
12-19-04.

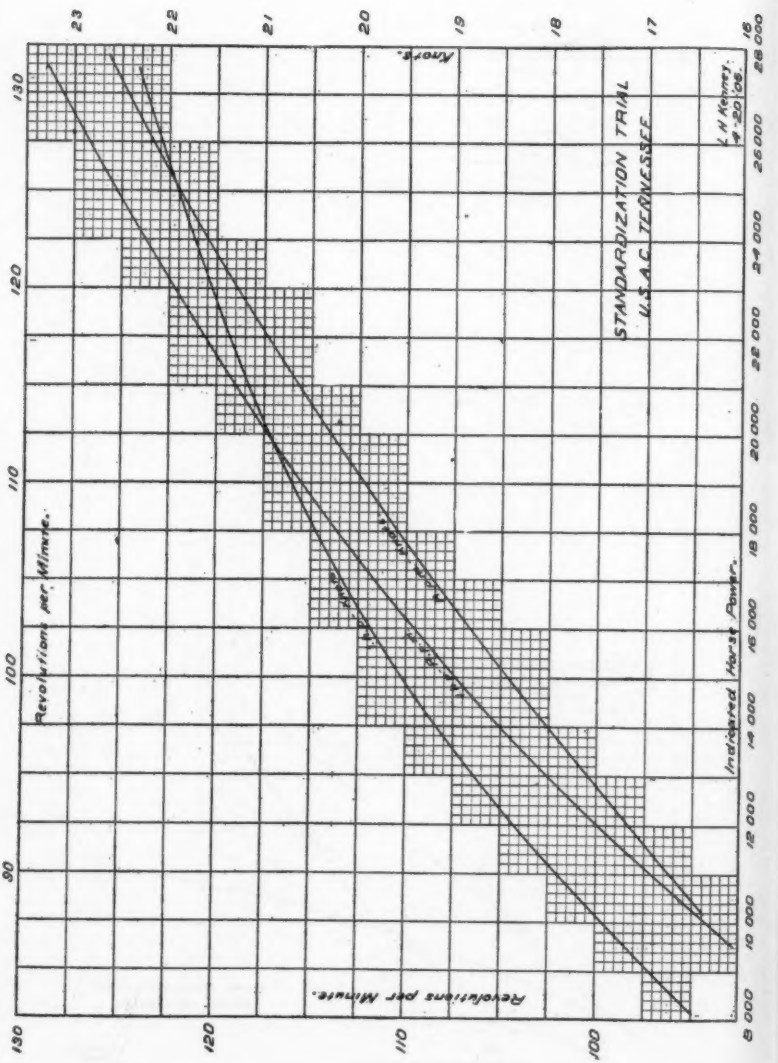

Note.
Discharge gauge reading 215 pounds, for test
Spring #100
Time 10.93
Card #1
Interpolation Formula
Symbols
I.H.P. = To be determined
I.h.p. = From curve
P = Discharge gauge reading
I.H.P. = .00564 I.h.p. P
Displacement
I.h.p.
MAIN FEED PUMP
U.S.A.C. TENNESSEE
L.H. Kenney
Double Strokes per Minute.
Water Piston Displacement in Cubic Feet per Minute
Indicated Horse Power.


Spring #100
Time 8.33
Card #1
Note
Discharge gauge reading 100 pounds, for test
Interpolation Formula
Symbols
I.H.P. = To be determined
I.h.p. = From curve
P = Discharge gauge reading
I.H.P. = .0056 I.h.p. P
Displacement
I.h.p.
I.H.P. Discharge Overboard
Note
Discharge overboard from bilge
FIRE AND BILGE PUMP
U.S.A.C. TENNESSEE
L.H. Kenney
Double Strokes per Minute.
Water Piston Displacement in Cubic Feet per Minute
Indicated Horse Power for Overboard Discharge
Indicated Horse Power.


Air Pressure in Inches of Water.
Revolutions per Minute.
Air Pressure.
I.H.P.
BLOWER ENGINE
U.S.A.C. TENNESSEE
L.H. Rowell
1-4-06
Indicated Horse Power
300
400
500
600


Spring #100.
H.P. Cylinder
Spring #40
L.P. Cylinder.
Cards taken at 1010 amperes
Note.
Volt meter reading 108. for rest.
I.H.P.
Kilowatts
Indicated Horse Power.
GENERATING SET No 1
U.S.A.C. TENNESSEE.
Amperes.
100
200
300
400
500
600
700
800
900
1000


Spring #160.
Time 0.30"
Card #2
Water Piston Displacement in Cubic Feet per Minute.
Inches in Inches.
Indicated Horse Power.
I.H.P.
DYNAMO CONDENSER PUMP
U.S.A.C. TENNESSEE.
Double Strokes per Minute.
20
30
40
50
60


STANDARDIZATION TRIAL
U.S.A.C. TENNESSEE
L.N. Kenny
4-20-08.
Knots.
23
22
21
20
19
18
17
16
Revolutions per Minute.
130
120
110
100
90
Indicated Horse Power.
8 000
10 000
12 000
14 000
16 000
18 000
20 000
22 000
24 000
26 000
28 000

METHODS OF CONDUCTING SPEED TRIALS.

By Naval Constructor J. J. Woodward, U. S. N.,
Member.

[Read at the thirteenth general meeting of the Society of Naval Architects and Marine Engineers, held in New York, November 16 and 17, 1905.]

I. Introduction.

The determination of the true speed of a vessel through the water is unfortunately one of those numerous problems within the field of Naval Architecture for which we have only approximate and not exact solutions. Yet this has by no means been due to a lack of interest in this important subject, or to any deficiency in either the number or the ingenuity of the methods which have at one time or another been devised for the purpose of accomplishing it.

It is not desired to attempt any historical review of these methods, however picturesque some of them may have been,—as for example, the device described by the Roman architect, Vetruvius Pollio, which consisted in a small water wheel secured to the side of the hull near the waterline, which as it was revolved by the movement of the vessel through the water drove a train of gears by means of which pebbles were slowly dropped, one by one, into a basin; so that by counting the pebbles the distance the vessel had run was known.

It will be necessary later to call your attention to one of these devices whose age, though not so great as that of the method proposed by Vetruvius, is still sufficient to inspire respect for its antiquity, even if we may not feel it for the accuracy of the results it furnishes. I refer to the method of measuring speeds by the common "chip log" which, in the form of a "current pole," has been extensively used for measuring the velocities of the tidal currents to which vessels were subjected while undergoing speed trials over long measured courses at sea.

Of the distinctly modern direct methods of obtaining the speed of a vessel through the water, the various forms of "patent" and electric logs, pressure logs, etc., which are of the first importance for purposes of general navigation, are entirely unsuitable for use in connection with speed trials, since their indications are so irregular in amount as to render them unfit for anything but approximate measurement.

And here it may be remarked that the essential difference between the measurements of the speeds of vessels by the patent log and by the propeller of the ship itself, is the fact that the entire power developed by the small propeller of the patent log is expended in overcoming the friction of its system and in the operation of its recording mechanism; whereas with the propeller of the ship, only a small part of the power developed is expended on the first and a wholly insignificant part on the second of these items.

As a result, the indications of the patent log are subject to sudden fluctuations of unknown amounts owing to changes in its internal friction; on the other hand, the ship's propeller, considered as a speed recorder, is only slightly affected by changes in internal friction of the machinery, and will give accurate comparative results, *provided* external conditions which affect the economy of propulsion, remain the same. That is, if the displacement, condition of the bottom, effect of the wind upon the vessel, etc., remain the same, then the number of revolutions of the ship's propeller permits an accurate determination of speed, provided we know accurately what speed has been developed over a measured base under exactly similar conditions. Without such similarity of external conditions, speeds based upon the revolutions of the ship's propeller are entirely fallacious.

A remark which should be made while considering the general subject of speed trials, is that the reports frequently published under this heading, descriptive of the performances of foreign naval vessels, are only "speed trials" in name, and are held primarily to test some quality other than speed. The speeds developed in such cases are simply data of general in-

terest, noted with as much accuracy as practicable, but as they have only an indirect bearing upon the definite object for which the trial is held, no special refinement of observation is necessary. Thus the eight-hour full-power trial held of all large vessels built for a certain foreign navy are conducted under contracts which require a certain stated I.H.P. to be developed by the engines for a given number of hours, but without any contract requirement as to the speed to be maintained. For such a trial, it may be entirely satisfactory to have the speed determined by runs in alternate directions over a twenty-mile course at sea, without any stake vessels to mark the course or to take current observations, and the nominal speed may be deduced directly from the elapsed times and the nominal distance run. But it is to be very much doubted if the speed so obtained, or as given by the readings of the patent log, is generally anything more than a crude approximation. Of course, since there is no financial penalty involved in such contracts for deficient speed, no real hardship is done the shipbuilder, but it seems at least undesirable to introduce into official records data obtained by such unreliable methods.

When, however, we consider the contracts for the vessels building for our own Navy, no such conditions obtain. Here, the principal contract trial is essentially one for speed, which is required to be maintained at sea continuously for a stated time, generally for four hours. The indicated horsepower required to obtain this speed is in no manner a part of the contract. With large vessels of high speed, in which when approaching their contract speeds small increments in speed result in a great corresponding increase in power, the determination, with the highest possible degree of accuracy, of the true speed maintained during the four-hour trial becomes a matter of the first importance. This will be readily realized, quite apart from the academic point of view of experimental research, when it is remembered that for a deficiency in speed the penalty provided for by the contract of certain ships now building is at the rate of $200,000 per knot for the first half knot below the contract speed, and at the rate of $400,000 per

knot if the speed falls more than half a knot below that required by the contract.

In view of the importance of the subject, it has been thought of interest to the Society to place before it the records of a number of recent speed trials as conducted over a long course at sea, so as to show the general character of the tidal-current corrections made for such trials. To this information has been added some considerations on the method of determining speeds at sea by first standardizing the screws over a measured-mile course.

Before, however, entering into a detailed description of the methods by which speeds are determined during contract trials, it seems pertinent to say a word upon the subject of the general character of such trials.

At the time most of the earlier steel ships of the Navy were designed, reciprocating triple-expansion engines and cylindrical boilers were in such general use as to form a practically universal type of machinery installation built on such well established lines that the real necessity for the trial was simply the determination whether the size of the boilers and engines was sufficient for the required speed and that the workmanship was satisfactory. Under these conditions the limitation of the contract trials to the single requirement of making a four-hour run at sea at full power was reasonably satisfactory as a proof that the vessel would be able to perform her service in a suitable manner.

With the introduction of water tubular boilers of various types, and especially at the present moment when the reciprocating engine is being displaced by various types of turbines, it is important that the contract trials should include, in addition to a run of a few hours at full power, such other trials as may be necessary to show that the endurance and economy of the machinery may be expected to be satisfactory in service.

As an illustration of the tendency in this direction the following outline is given of the contract trial requirements for recent vessels ordered for our own and a foreign naval service:—

Recent contract trial requirements of U. S. Naval Vessels.

I. For a recent battleship,
  (a) a progressive speed trial over a measured mile course.
  (b) a speed trial at sea of 4 hours' duration at full power.
  (c) an endurance trial of 24 hours' duration at sea at not less than ⅔ the I.H.P. developed during the 4-hour trial, the coal consumed to be accurately measured, but no penalty for excessive coal consumption.

II. For a fast cruiser of small size,
  (a) a progressive speed trial over a measured-mile course.
  (b) a speed trial of 4 hours' duration at full power.
  (c) a 24-hour endurance and coal-consumption trial at a specified speed (about .9 full speed). A money penalty is enforced in case the coal consumption exceeds a specified amount per knot run by the vessel.

Recent requirements of British Naval Vessels.*

I. Battleships, first-class cruisers, cruisers, second-class cruisers.
  (a) Preliminary trial at sea.
  (b) 30-hour trial at about one-fifth full power. This is a trial for coal consumption, and any excess in consumption above a specified amount per I.H.P. per hour results in applying a penalty.
  (c) 30-hour trial at about 70 to 75 per cent. full power. This is essentially an endurance trial, and excessive coal consumption results in no penalty.
  (d) 8-hour trial at full power. The contract requirement is horsepower and not speed. Any deficiency in power results in applying a penalty.
  (e) Trial after opening up (24 hours).

II. Third-class cruisers.
  (a) Preliminary trial at sea.

*E. L. Altwood, R. C. N. C., "War Ships"— London, 1904.

(d) a 24-hour endurance and coal-consumption trial at a specified speed (about ½ full speed). The coal to be weighed, but no penalty for excessive coal consumption.

(b) 30-hour trial at half forced-draft power.
(c) 8-hour trial at authorized natural-draft power.
(d) 4-hour trial at full forced-draft power.
(e) Trial after opening up (24 hours).

It will be seen that these requirements necessitate the vessels undergoing trial to be run a number of days at sea under widely different conditions of operation. Moreover, the present tendency in the determination of the detail requirements for such trials is towards conducting them as nearly as may be under service conditions. Full particulars of recent modifications in trial requirements for British naval vessels, are given in the London "Engineering" of November 11, 1904. These modifications limit the number of men that can be employed in the engineer and fireroom force; limit the amount of lubricant that may be used, and forbid its application in any manner other than through the system of lubrication regularly provided; and finally limit the clearances of the principal assemblages to good practice, forbidding them to be greater than specified amounts.

The point to which it is desired to call particular attention is the fact that in our own service, these elaborate and extended trials are required to be carried out by the shipbuilder who contracts to build the vessel, with a crew employed by him for that particular occasion.

Such a method of procedure appears to be extravagantly wasteful, not only of money but of effort that should more profitably be directed in other directions. So far as I know, our own service is the only one in which such a practice is maintained, and it is thought that a change in this matter would be beneficial in the highest degree. If a vessel when entirely completed by her builders was put in commission with a special naval trial crew, the only employees of the contractor on board

being such leading men as were necessary for the builder to still retain responsible direction of the operation of the machinery, it is believed that a proper division of responsibility for results could still be retained, while vessels valued at millions of dollars would not be placed in the hands of an extemporized crew having no responsibility for the future efficiency and endurance of the machinery. Such a method of procedure cannot fail to result not only in a more rational method of conducting such trials, but in a longer life for the machinery that is preserved from trial under present conditions.

Returning to a consideration of the methods of conducting speed trials under present conditions, consider first the method of determining speed by a double run over a long measured course established at sea.

II. Description of Track Followed by Vessels Making Speed Trials Over a Long Measured Course at Sea.

The measured course which has generally been selected on the Atlantic Coast for the purpose of conducting speed trials at sea of large vessels of high speed is located off the New England coast. A track chart of this course is shown in Plate I. In this plate are shown the ranges and the principal angles by which the buoys are located that mark the various stations establishing the course.

The southern end of this course is to the eastward of Cape Ann and is marked by a buoy the position of which is accurately determined by a range and by angles between other objects none of which are more than eight miles distant. Owing to its proximity to the Cape, this trial course is generally referred to as the Cape Ann Trial Course.

The course then proceeds on the bearing N. 22° 6′ E. (magnetic), its direction being marked every 6.6 nautical miles by buoys for a distance of 33 miles. At this point, the position of the buoy (station No. 6 in Plate I) is also determined by a range and angles between other objects, but the distances of

the latter are greater than at station No. 1, being in some instances as much as twelve miles from the buoy.

For fast vessels, such as the 22-knot armored cruisers of the *Colorado* class, which require a trial course more than 33 miles in length, the course just described is extended eleven miles more to the north, making a total length of 44 miles. The locations of the two buoys marking this extended portion of the course, stations Nos. 7 and 8, as well as those at stations Nos. 2, 3, 4 and 5, are all determined solely by angles between objects, some of which are as much as 16 miles distant; and in some cases, as for the buoy at station No. 3, the character of the angles is by no means the best for an accurate placing of the buoy. This is not a serious drawback when such buoys only mark the direction of the course for a vessel making the full 33-mile run. On the other hand, when the distance betwen stations Nos. 1 and 3 is taken for the trial course of a slow vessel, as was done in the speed trial of the monitor *Nevada,* any error in locating the buoy at station No. 3 directly affects the speed obtained.

An examination of the data obtained on the speed trial of the *Nevada,* as given in Table I, and its comparison with the results of trials for other monitors of the same design, tends to show the possibility that an error was made in determining the speed of this vessel, and that this was due to buoy No. 3 being out of position during the trial.

It is to be noted, moreover, that even in the case of the ranges which aid in the accurate location of the buoys at the end stations Nos. 1 and 6, these ranges are not at right angles to the direction of the trial course, and, consequently, a buoy may be exactly on the range, yet materially out of position, as affecting the length of the course, if not anchored so that the angle between the range and a distant object is correct.

Now, the matter of the maintenance of the trial course buoys in good order requires them to be lifted from time to time, overhauled at a shore station and then replaced. Heavy weather, and especially field ice in the winter, are possible

causes of displacement of the buoys from their correct positions.

As a result of the above conditions, it is necessary, as soon as the date of trial of a new ship is determined upon, to have a vessel proceed to the trial course and, at each buoy, verify the angles between the shore beacons so that if not correct the positions of the buoys may be rectified. Frequently, the interval between the date of fixing the time of trial and the trial itself is short, so that it is impossible to wait for the most favorable conditions of atmosphere to take the necessary angles. Now, measurement with a sextant of an angle between two objects is a simple enough operation if the objects can be distinctly seen; but when one of them is a beacon 12 to 20 miles away that is not visible at all under unfavorable weather conditions, there is at least a possibility of error when the work of verification of the positions of the buoys must be completed within a limited time.

Moreover, the vessel from which these observations are taken must be fitted so as to be able to raise the buoys with their chain and anchor, so that if any buoy is found away from its proper position it may be replaced at once. This necessitates the use of a lighthouse tender, or other specially equipped vessel, and even with such a vessel, the handling and accurately anchoring, in their predetermined positions, of buoys of this large size in so great a depth of water, in anything but a perfectly smooth sea, is something that must be performed before its difficulty is fully appreciated.

The above summary description of the preliminary precautions necessary to be taken to ensure accuracy in the length of the trial course are sufficient to show the general character and amount of work involved.

The depth of water on the Cape Ann Trial Course is generally 40 fathoms or more, except that near station No. 2 there are two lumps leaving only 25 fathoms, and between stations Nos. 7 and 8 the depth decreases from about 40 to 30 fathoms. Soundings at different points of the track are shown on Plate I.

III. Method of Conducting Speed Trials Over a Long Measured Course at Sea.

Passing now to the details of the method of conducting a speed trial over such a course, the first step is to obtain the necessary number of observing vessels to act as stake boats and to take current and other observations during the trial.

It is important that these stake boats should be of such a size as to be plainly visible under all ordinary conditions of the atmosphere, at or before the time that the vessel undergoing trial passes the preceding stake boat, so that the steersman may be assisted in keeping a straight course.

For this reason, as well as for considerations mentioned later in connection with current observations, it is important that the stake vessels should be of large size. Tugs, converted yachts, and torpedo-boat destroyers are not satisfactory when used for this purpose.

Written instructions (see Appendix 1) are issued to each stake boat, which direct it to anchor about two hundred yards west of the buoy marking its station and to avoid either fouling the buoy or masking it by tailing across the line. As the stake boat is seen some time before the station buoy is visible, if the former is not in her proper position the buoy will not be sighted where anticipated and the consequent change in course to return to the true track of the trial course results in an increase of distance traveled which it is difficult to properly allow for. At times, the errors from this general cause have been so important that some approximate allowance had to be made for them.

Thus on the trial of a small cruiser in the run southward after leaving the second stake boat, the third stake boat (a torpedo-boat destroyer) could not be seen owing to a light fog bank. The cruiser mistook a fishing smack for the destroyer and ran off a point to the westward, so that when the destroyer was sighted about the middle of the distance between the two stations, she bore on the port bow of the cruiser. The allowance made for this cause (see Fig. 1, Plate VII) was 0.1292 nautical miles additional distance run.

In the trial of the same cruiser a further allowance of 0.0356 miles additional distance run was made owing to a fishing schooner crossing the track and obliging the cruiser to run two points off her course for one and a half minutes to avoid a collision.

The sum of the corrections of the distance run owing to these two incidents brought the corrected speed of the vessel from 16.40925 knots to 16.4506 knots. While the amount of this correction is small, and is not liable in similar cases to be very much greater, yet the circumstances of obtaining the data by which such corrections are made are of a very undesirable character to introduce into a contract trial for which there is a financial penalty in the case of deficient speed. This is due to the fact that the change in course under such circumstances is usually made immediately on discovering the necessity therefor, and its value, as well as the exact positions of the vessel between the adjacent stake boats when the change in course is made, are necessarily estimates made after the event.

IV. Instructions for Current Observations.

The stake boats marking the course also act as observing vessels to note, during the day the trial is held, the strength and direction of set of the current, the direction and force of the wind, and the character of the sea. An extract from the instructions in accordance with which these observations are taken, and the form in which they are recorded, is given in Appendix I.

The principal feature in these instructions to which it is desired to draw attention are the instructions for taking the current observations at 15-minute intervals. It will be seen that a large amount of stray line (21 fathoms, or 126 feet, in all) is provided, so that the current pole may be well clear of the ship before the current measurement begins. In the earlier trials held over this course, the current was measured by the amount of line paid out in 28 seconds, the time being taken by a 28-second glass. This resulted in very small lengths of line being paid out,—for example, with a quarter knot per hour

current it was less than twelve feet, or less than one tenth of the stray line. In an endeavor to obtain greater accuracy in the current measurements, the instructions were modified to their present form, by which the number of seconds necessary for a given length of line (usually 94 feet and 6 inches) to be paid out are measured with a watch, or, if this time is in excess of five minutes, the length of line paid out in five minutes is noted.

All current observations are taken by a commissioned officer who signs the report, certifying to the accuracy of the results given therein; and special emphasis is placed in the instructions to commanding officers of observing vessels, upon the necessity of the vessels being motionless when the tide is measured. The tidal records given in this paper have all beeen taken under these instructions and they were doubtless carried out as far as practicable. The impossibility of keeping a vessel motionless that is anchored in the open sea in from 30 to 60 fathoms of water, while observations are made which require about five minutes to take the necessary measurements for each observation, and are repeated every 15 minutes, is however sufficiently obvious to require little comment.

The importance of having large ships for observation vessels, so that they will lie more quietly at their anchorage than tugs, torpedo-boat destroyers, etc., has been repeatedly shown. On the trial of a certain battleship, all the observation vessels were tugs, and the Trial Board in its report called attention to the necessity of much larger vessels being employed as stake boats.

Whether more accurate current observations can be obtained from a boat moored to the station buoy, when there is enough breeze to swing the observation vessel at her anchorage, and in anything but a perfectly smooth sea, is, I think, very much open to doubt.

It must further be remembered that, although the observers measuring the current are commissioned officers, it is probable that none of them have had much, if any, practical service experience with the "chip log," which has passed away in the naval service as an instrument of general use in practical navi-

gation almost as completely as the automatic recording log of Vetruvius. Moreover, these observers are never the same on two different trials. Now, the fundamental feature governing accuracy of observation with the current pole is good judgment in maintaining a proper tension in the cod line by which the drift of the pole is measured. If kept too taut, the pole will tend to be drawn home and the current be underestimated; if too slack, the fact that the line does not lay perfectly straight when in the water, but tends to take a more or less sinuous form, will result in overpaying and the current be overestimated.

Commander Lecky in his book "Wrinkles in Practical Navigation" (9th edition, page 177) says: "logs, whether patent or common, are entirely unsatisfactory in their indications of speed." I think the same thing may be said of the measurement of current velocities by the pole and line.

In concluding the examination of this particular phase of the subject I think it only fair to add, that in conversation with officers who have taken these current observations, I have never heard anyone express a belief in the real accuracy of the current velocities obtained, no matter how painstaking the collection of the data; whereas, I have heard a most careful and experienced officer state that after taking special precautions to ensure accuracy in his observations, he felt no confidence in the results being more than crude approximations; and in this case the stake boat was a battleship and the observations were taken under the most favorable circumstances.

V. Details Relating to the Conduct of the Trial.

Passing now to the conduct of the trial itself. This simply consists in the vessel undergoing trial making one double run over the course. Beginning usually at the southern end, the vessel runs to the north, and at the precise instant when each of the buoys marking the stations of the track bear abeam, the time is noted. This observation is made by sighting across the edges of two vertical battens which are placed in a plane at right angles to the axis of the ship, and noting the time with

a chronometer when each buoy passes the line of sight.

After terminating the run to the northward, the vessel makes a turn with a small helm angle, being brought to a proper heading for returning over the course at some distance before reaching the northern station, so that she may have acquired full speed after completing her turn and before passing this buoy. During the turn, the throttles on the main steam pipes are not allowed to be touched and no alteration is permitted to be made in the manner of operating the machinery; but the performance of the vessel during the turn is not considered to be a part of the trial.

The return run is then made to the southward, the time being taken as before when each station buoy is passed.

VI. Correction for Currents Observed on the Trial Course.

The principle upon which the observed speeds are corrected for the effect of the tidal currents, is to admit that the sphere of influence within which the strength of the current as observed at any stake boat extends, is from a point half way between it and the stake boat to the north to a point half way between it and the stake boat to the south. That is to say, in beginning a run to the northward, it is admitted that the current experienced by the vessel throughout the portion of the track from station No. 1 to a point half way between stations Nos. 1 and 2, is uniform both as regards direction and intensity while the vessel is passing over this portion of the track; and that its value equals that of the mean value of the current measured by the observation vessel at station No. 1, both as to direction and velocity for the period of time that the vessel undergoing trial required to traverse this portion of her trial run. The portion of the trial course just described is referred to as "Square" No. 1 and is so indicated on the chart of the trial course shown in Plate I.

In a similar manner "Square" No. 2 is that portion of the course extending from a point half way between Stations Nos. 1 and 2 to a point half way between Stations Nos. 2 and 3;

and so on for the rest of the course; the whole course being thus divided up into "Squares," within each of which the speed of the vessel is affected, respectively, by the mean value of the current observed at the station vessel in the center of the "Square."

On the return run from the north to the south, the same division of the course into "Squares" is preserved but the numbers assigned to them (for purposes of identification in the calculations for current corrections) form a continuous sequence with those designating the "Squares" on the run north. (See Table II giving the result of current correction in the different "Squares" for the official speed trial of the U. S. Battleship *Missouri*.) The track chart shown in Plate I has the numbers of the "Squares" marked for the runs northward only.

The method of correcting for the effect of the current within a given "Square" is as follows: Let AB in Fig. 2, Plate VII, be the distance in Square No. 1, measured on the true course between Stations Nos. 1 and 2, the point B falling exactly half way between these two Stations. For the time that the vessel is going from A to B, the mean value of the strength of current and its mean direction are estimated, based upon the current observations taken by the observing vessel stationed at A. Let—

T=time the vessel is in the "Square."

D=distance AB in the "Square," measured on the true course.

c=amount of the current developed during the interval of time T.

x=the angle of the keel of a vessel with the direction of the current.

y=the angle between the compass bearing of the ship's head (as necessary to overcome the effect of the current) and the compass bearing of the true course.

An inspection of Fig. 2, Plate VII will show that the corrected distance which the vessel has steamed through the water

while within the "Square" is $A'A$ and that it may be calculated in the following manner—the numerical figures being taken in this instance from the Current Correction Sheet of the *Missouri's* trial, Table II. Attention should be called to the fact that the algebraic sign of the quantity a depends upon that of cos x, which varies according to its quandrant (in the particular case in question cos x is negative).

SQUARE NO. I.

$D = 3.3$

$c = 0.0745$

$x = 142^\circ\ 0'$ (cos negative).

log. cos. $x = 9.89653$ log. sin. $x = 9.78934$

log. $c = 8.89487$ log. $c = 8.89487$

$a = -0.06186$ log. $a = 8.79140$

$b = 0.04833$ log. $b = 8.68421$

$D - a = 3.36186$ (Subtract) log. $(D-a) = 0.52658$

$y = 0^\circ\ 49'$ log. tan. $y = 8.15763$

log. $(D-a) = 0.52658$

(Subtract) log. cos. $y = 9.99996$

$A'A = 3.3622$ log. $A'A = 0.52662$

In a similar manner each of the corrected distances that the vessel has actually run through the water in each of the "Squares" during the run north, are calculated and the sum of these distances and the sum of the elapsed times between the station buoys permit the true speed to be calculated for the run north.

On returning, similar corrections are made during the run

south; and finally, the total sum of all the corrected distances is divided by the sum of all elapsed times in order to obtain the final speed for the trial. Table II shows this work in the determination of the speed during the trial of the *Missouri*.

It will be seen that this method of correction for the current makes full allowance for all set of the current across the line of course, but that it presumes abrupt changes of tide at the points where the vessel passes from one "Square" to another. While, of course, the latter hypothesis is necessarily incorrect, it does not seem practicable to make any more satisfactory assumption.

VII. Examination of Current Data as Reported on Certain Trials.

The most unsatisfactory feature in connection with the current corrections is, however, the general character of the measurements of the strength of the current. To illustrate this graphically, curves have been prepared showing the strength of current at each observing vessel during the trials of the *Missouri, Denver, Colorado, West Virginia* and *Pennsylvania* (Plates II to VI inclusive*). These particular vessels were taken at hazard as samples amongst recent trials for which the data were most conveniently available, with the exception of the *Missouri* and *Denver,* which were selected because their trials occurred on two successive days when it would have been expected that there should have been some general agreement between the direction and strength of current at corresponding stages of the tide. The times of high water at the Boston Navy Yard, which is the reference point for tidal data over the portion of the coast within which the trial course falls, are marked on the various sheets.

The strength of the current and its direction relative to the magnetic bearing of the trial course are indicated for each of the station vessels, and the direction of the wind relative to the course and its force by Beaufort scale are also shown. The condition of the sea is also stated at the time the vessel undergoing trial passed each station vessel.

* Plates IV to VI are omitted in this reprint, as the unsatisfactory character of the current observations are sufficiently well shown by Plates II and III.

A comparison of these records in the case of the trials of the *Missouri* and *Denver* shows that although when the *Denver* was tried, the time of high water at Boston was only 48 minutes later than on the *Missouri's* trial, yet there was almost a complete reversal of the direction of current in the majority of instances.

It has sometimes been assumed that the strength of the current on this trial course was so small that its effect on the speed of the vessel was insignificant, and, therefore, that accuracy in the current observations was unimportant, since the errors in these small quantities would approximately compensate for each other, that in any case their amount was trivial, and hence the question of current compensation in a double run over this course was unimportant.

An examination of the current curves shown on the accompanying plates does not indicate this to be the case, at least so far as the reported strength of current is concerned, observations of more than one knot per hour being at times recorded.

In Table III is given a summary of the principal results of the trials of the five vessels referred to above; and if we compare for each of these vessels, the speed corrected for the effect of current with the uncorrected speed, it will be seen that the difference between the two varies from nothing to 0.18 of a knot, and is indifferently positive or negative. In the case of the trial of another battleship, which is not included in this Table, the observed times between the end Stations gave:—

Speed uncorrected, run north	17.3584 knots.
Speed uncorrected, run south	18.5727 knots.
Difference	1.2243 knots.

In this trial the current correction applied to the observed speed during the run north, was nearly four-tenths of a knot for the average during that run. It must be realized, therefore, that the errors in the current observations, far from being negligible, vitally affect the accuracy of the final result in the speed so determined.

It should be further noted that on account of the length of the trial course, its distance from Boston and the necessity of making the run in daylight, the time of actually beginning a trial run over the Cape Ann course, is practically independent of the stage of the tide or of the force and direction of the tidal current. Conditions in this respect must simply be taken as found, without any real latitude of choice. Any nominal attempt to so choose the time of beginning the run over the course as to equalize the action of the current must, therefore, be regarded as simply a verbal form, the only result of which is to obscure the real facts through a reliance upon fancied conditions, which do not and cannot exist.

Under the head of current observations during measured-mile trials, further reference will be made to the probable accuracy of the velocities obtained with the current-pole outfit, but from an examination of the current curves shown on Plates II to VI, on their very face, the manifest irregularities and contradictions in the records appear to be enough to stamp this method of trial as unreliable for any purpose and especially to be avoided when large financial issues in connection with the possible enforcing of penalties due to deficient speed may depend upon the results obtained by it.

VIII. General Theory of the Determination of a Vessel's Speed at Sea by Means of Runs Over a Measured-Mile Course.

To determine the speed of a vessel during a trial at sea of several hours' duration by means of a knowledge of the number of revolutions per minute made by her propeller, a series of runs is first made over a carefully measured trial course of moderate length (usually taken as one nautical mile). For each run the speed of the ship, the revolutions of the propeller, and the corresponding I.H.P. of the main engines (together with such part of the auxiliary machinery as contributes to propulsion) are all carefully noted.

While the data as to horsepower do not directly assist in the determination of speed, it is absolutely necessary that they

should be collected, if a proper analysis is to be made of the efficiency of propulsion of the system consisting in the ship, its engine and propeller; and the accurate calculation of the power developed by all machinery that contributes to propulsion should be considered as essential a part of a properly conducted speed trial as is the record of the number of revolutions of the propeller. Especially in the case of a failure of a vessel to attain her designed speed, as required by contract to be developed on trial, does a complete record of I.H.P. and other engine-room data become a matter of the first importance as assisting in the discovery of the particular defects to which an inadequate performance is due.

Runs are therefore made by the vessel over the measured-mile course in alternate directions with and against the tide so that, by methods to be examined later, a curve may be constructed that represents empirically the relation between the number of revolutions per minute of the propeller and the corresponding true speed of the vessel through the water.

Having prepared this curve, the vessel undergoing trial is then taken to sea and subjected to a trial of the number of hours duration required by the contract for her construction. During this sea trial the shaft counter is read at regular intervals, usually every 15 minutes, and the average number of revolutions per minute of the propeller is determined for each of these periods. The true speed corresponding to each of these numbers of revolutions is then taken from the curve of speed revolutions, and the speed for the entire trial determined by taking the average of the speeds during the 15-minute periods.

In general principle the method of conducting such a trial is simplicity itself; but in practice there are many difficulties of the character that are usually encountered when any attempt is made to make exact physical measurements. Moreover, in the conduct of such standardization trials, as in the case of any other method of conducting contract trials, the conflicting interests of the two contracting parties have to be equitably reconciled, so that, on the one hand, the speed of the vessel

may be accurately known, and on the other, that the ship-builder may be put to no expense in the conduct of the trial beyond that necessary to demonstrate the fact that the contract for the construction of the vessel has been fully complied with.

The first step in preparing to properly conduct such a trial, is the selection of a location where a suitable trial course may be established. And here the primary necessity is clearly that of having a sufficient depth of water on the course to permit the highest speeds to be developed without any appreciable interference to the wave systems established by the vessel owing to the proximity of the bottom. This applies not only to the measured-mile course itself, but to the depth of water on the approaches to the course, within which the vessel should acquire the full speed which is to be measured between the ranges marking the mile. The breadth of the deep-water channel should also be ample, and the buoys marking the trial course should not be located close to the edge of relatively shoal water.

It is to be regretted that when we pass to quantitative considerations, the data available are all too meager. Among recent papers relative to this subject may be mentioned those by Mr. Harold Yarrow and Mr. W. W. Marriner read at this year's summer meeting of the Institution of Naval Architects, and a most interesting one by Naval Constructor Paulus, I. G. N., published in the "Zeitschrift des Vereines Deutscher Ingenieure" of December 10, 1904.

All of the trials described in these papers relate to vessels of the torpedo-boat destroyer class, and most unfortunately experimental data are lacking that confirm the inference that the depths which have been found satisfactory for these small vessels at certain speeds, are practically equally so for much larger vessels at the same speeds. It is greatly to be regretted that the results of trials of one of our own high-speed vessels of large displacement over our own Provincetown, Mass., and Rockland, Maine, measured-mile courses are not available, so as to permit this question to be definitely settled—at least, for the courses experimented with and up to speeds of 22

knots. I trust at some later date it may be possible to lay before the Society the results of such comparative trials.

Next in importance to depth of water is the character and direction, relative to the course, of the tidal currents which will be met during the trials. The most undesirable feature in the current is *irregularity*, for, as we shall see, a regular current can be accurately allowed for, but irregular fluctuations inevitably escape analysis, and they are, in my opinion, the most important source of error in this method of determining speed. As the times of high and low water usually differ considerably from the times of slack water it is desirable that a careful investigation, extending at least over one lunar month, should be made of the currents existing on any trial course which it is proposed to use for high-speed vessels of large displacement. While such observations can only lead to average results, which are changed to a greater or less extent by exceptional conditions of weather, such as strong winds lasting for a considerable length of time from the same point of the compass, yet information as to average conditions would afford a means of approximating the time when the tidal currents might be expected to be most regular in their rate of change. In the absence of such information, and very possibly to supplement it if it were available, a vessel should be anchored as near the middle of the measured-mile course as practicable, with the same arrangements to take current observations as have been described for the observing vessels on the long sea course. These observations should preferably *begin the day before* the standardization trial and continue uninterruptedly until that trial is over. By this means it would be possible on the morning of the standardization trial to estimate, as closely as may be done, the probable time of slack water occurring during or nearest to the trial and thus avoid making any of the runs at the more important speeds (viz. those that approximate the contract speed of the vessel being tried) at the stage of the tide when irregularities in the tidal currents are the most likely to occur.

It need hardly be said that, in so far as the construction of

the curve of true speeds and revolutions is concerned, no direct use is made of the current observations taken from the anchored vessel. Such observations of the strength of current are, however, of great value in forming an opinion of the general character of the accuracy with which the strength of a current can be measured by the current pole and line as has been customary on the Cape Ann Trial Course. In Figs. 1, 2, 3 and 4, Plate X, are given the results of a number of such observations taken from anchored vessels, while another vessel was making runs over a measured-mile course. The strength of current as plotted is not its full strength, but is that of the component of the total velocity of the current as projected on the direction of the length of the trial course. In comparison with these curves of measured strength of current are shown other curves, the ordinates of which are deduced from the results of runs with and against the tide made during the standardization trial itself. An examination of these curves will, I believe, go far to show the character of accuracy (or inaccuracy) that may be expected from the current-pole measurements. But, while unsatisfactory quantitatively, the current curves from direct measurement show with sufficient accuracy the periods of irregularity in the force of the current and when the change from flood to ebb occurs; if such curves were available for the twenty-four hours preceding a standardization trial, they would assist materially in the proper timing of the high-speed runs so as to insure their being made at a suitable stage of the tidal currents.

In the matter of the direction of the set of the current, relative to the bearing of the trial course, while, of course, it is preferable to have the current exactly in the direction of the course, it is doubtful if a moderate angle between the two is a serious disadvantage, *provided* that while the vessel is running over the mile no attempt is made to keep her on an exact line of bearing but simply to maintain a steady compass course. Under these conditions the component of the current at right angles to the course simply sets the vessel towards or from the trial course buoys and does not affect the speed of crossing the

ranges marking the beginning and end of the mile. It is possible that accuracy in steering under these circumstances might be materially facilitated by the use of a special compass card of more than standard diameter.

Finally, the measured-mile course should be so selected that it may be possible to establish suitable beacons marking the ranges and locate them in proper positions. This is not as simple a matter as it may seem at first sight, for if the land rises rapidly from the water's edge it may be impracticable to make the distance between the front and back beacons of a range a suitable amount, without having a front beacon of such a height as to render its cost of construction prohibitive. This at once raises the question as to what constitutes a proper distance between the beacons. Professor Peabody, in his "Naval Architecture" (page 528), recommends for a trial course 500 yards from the shore, a distance between the beacons of 300 yards.

This would be entirely satisfactory, but under certain circumstances even a less distance between the beacons could be accepted as giving sufficiently accurate results. In determining what is the least distance between the beacons marking a range, in order that they may give satisfactory results, it is first necessary to assume some permissible maximum error in the speed due to this course. Take one hundredth of a knot as the value of this permissible error; it follows that for a trial speed of 20 knots per hour we must know the length of our mile course within three feet—that is to say, the actual distance apart of the two positions of the observer at the respective instants when he believes he is crossing the ranges marking the end of the mile course should not exceed a true nautical mile by more than three feet or fall below a true mile by more than the same amount. It is evident that if at each end of the mile the beacons marking the ranges can be so designed and located that the maximum error in the position of the observer, when making his observation for time at the supposed instant of passage of each range, does not exceed one-half this amount, the requirement as to the permissible error will be fulfilled.

Consider now just what happens when the observer is about to cross the line of a range. A properly constructed beacon (see plate XV) should have a large triangular base permitting it to be readily picked out amongst the objects in the landscape and be surmounted by a pole of the smallest diameter that is visible from the ship, so that the apparent transit of the image of one pole before another defines the instant of passing the range. As the observer on the vessel approaches the range, the apparent opening between the poles of the front and back beacons decreases to zero at the precise instant of the commencement of occultation—*i. e.*, when the line of sight from the observer passes tangent to one side of the pole of the front beacon and to the opposite side of the pole of the back beacon. This is really the instant which should be marked with the stop-watch or chronograph, as it corresponds to a definite perception. With properly designed ranges, in which the ratio of the distance from the ship to the front beacon to that between the two beacons is practically the same for each range, no error whatever is made by this method of marking, and it is probable that in practice this is what generally occurs, even although the mental process may not be recognized. The probability that this is the case may be shown by the following considerations. The Carl Zeiss Optical Company, of Jena, assume in their description for using their stereoscopic range finder, that 30″ is the limiting value of the perception of depth with free vision. At the distance of one mile, 10.6 inches would measure an arc of this angular value. Now consider the appearance of a back beacon pole of suitable* size for the proper marking of time, just when the occultation of the back by the front beacon begins. At this time a pole 3⅞″ square and 7,800 feet distant will cover an angle of about 8½″ of arc. It appears, therefore, extremely doubtful, especially at high speeds, whether the eye is capable of performing any act of judgment that requires the separation of perceptions relating to the front and back beacons respectively during the actual

*A beacon having a pole made of a 8⅛ inch square timber, has given satisfactory results as to visibility, the beacon being 7,800 feet from the buoy marking the course.

period of occultation—such for example, as estimating when the center line of the pole of the back beacon is exactly in line with that of the pole of the front beacon. It would seem probable therefore that the only sensation which the eye can accurately perform in this connection, is the determination of the instant when occultation begins, and that to attempt to estimate by the eye the mid-instant of the period of occultation is impracticable. Since the attempt to estimate this instant of centering of the beacons will probably be made, it is important that the error from this cause should be reduced to a minimum, and returning, therefore, to the matter of the proportions of the parts of a range in order that the error in speed may not exceed a given amount and referring to Fig. 3, Plate VII, let—

a = the distance between the two beacons of a range.

b = the distance from the ship to the front beacon.

t = the diameter of each of the two beacon poles.

x = intercept on the line of the course corresponding to the entire period of partial or total occultation of the back by the front beacon.

Then, assuming that quantities of the order $\frac{t}{a}$ may be neglected, we have—

$$x = t\left\{1 + \frac{2b}{a}\right\}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

or,

$$\frac{b}{a} = \frac{1}{2t}(x - t). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

It follows from the above that, if we wish to have the possible error due to the design of the ranges less than one-hundredth of a knot up to a speed of 20 knots, in which case $x \leq 36$ inches, and if the distance of the trial course from the shore is such that the least permissible diameter t of the beacon poles is 6 inches, in order to be properly visible, then the maximum value of $\frac{b}{a}$ would be

$$\frac{b}{a} = \frac{1}{2 \times 6}(36 - 6) = 2\tfrac{1}{2}.$$

In the case cited by Professor Peabody, a pole 3 inches in diameter or less would probably be visible if the course was only 500 yards from the front beacon, in which case

$$a = \frac{2bt}{x - t} = \frac{2 \times 500 \times 3}{36 - 3} = \frac{3000}{33} = 90.9 \text{ yards.}$$

This, of course, is the inferior limit, as it is evident that in case the pole of the back beacon remains plainly visible, the *greater* the distance between the two beacons the better.

It is moreover evident that if we consider both ranges of a measured-mile course, Fig. 4, Plate VIII, the period of partial or total occultation should not only be sufficiently short but that also the ratios $\frac{BC}{AB}$ and $\frac{EF}{DE}$ should be as nearly equal as practicable. There is one reason for this equality in addition to those previously advanced, that is the importance of having everything contribute to maintaining the personal error of the observer the same when marking time on the two ranges. If the range at one end of a course is much "quicker" than that at the other end, owing to the time of occultation during the transit across the range being much less in one case than in the other, it appears inevitable that the different impressions conveyed by the two phenomena, will tend to produce a difference in the personal error of the observer. Fig. 4 gives the approximate distance between the beacons and the trial course at Provincetown, Mass., and is a good example of a course that might be improved by a better location of the back range marks.

The color of the beacon tripods and poles should also receive attention. Black and white stripes for the lower tripod and similar alternating bands of color for the poles would appear to be suitable for varying conditions of background and sky. The back beacon of the East Range at Provincetown, Fig. 1, Plate XIV, is a pillar between two windows in a church steeple. The pillar and all adjacent parts of the steeple are painted the same color, which at the distance of a couple of miles adds materially to the difficulty of distinguishing it. Marks of this character are very unsatisfactory for accurate work.

In Figs. 1, 2, 3, 4 and 5, Plate XIV, are shown photographs of beacons on some of the ranges at Rockland and Provincetown. That shown in Fig. 2 is satisfactory in general form, but those shown in Figs. 4 and 5 are very unsatisfactory, not only because their form and dimensions are not such as to give a quick, distinct transit of one beacon across the other, but also because, the flat board faces of the beacons being in a nearly north and south plane, when the sun is shining the face towards the ship is thrown in shadow after noon, and the beacons then become very difficult to distinguish from neighboring objects. The shape of both the pole and tripod of a beacon should be such as to be well lighted under all positions of the sun in which observations can be made. The pole should preferably have a circular section and the tripod be a pyramid with equilateral base. Flat shapes should always be avoided.

Whenever possible the poles of the beacons should show against the sky line. Where this is not possible, and the beacons are located at a great distance from the trial course, it may be necessary to place an artificial background behind the back beacon so that the images of the two beacons may fall on a background of a desired color.

IX. Calculating the True Speed from the Speeds Observed on the Measured Mile.

In approaching a discussion of the method by which we obtain the true speed of a vessel through the water by means of observations taken during runs over a measured course of known length, it will, perhaps, facilitate a presentation of the principles involved if we first consider a simple numerical example.

In Fig. 6, Plate VII, let *OB*, represent the axis of time and *OA*, the axis of current velocity. Suppose that at the time 1h. 00m. which is represented by the point *C*, on the axis of time, that the velocity of the current is at the rate of 0.2 of a knot per hour and that the current increases uniformily with the elapsed time at the rate of 0.1 of a knot in every ten

minutes. The curve of current velocities will, therefore, be represented by a line *DE*, during an interval of time within which the current increases at this uniform rate. Suppose that this current exists on a measured-mile course and flows in the exact direction of the length of the course, and without any change in direction during the period of time considered.

Let a vessel be now run over this course, with her engines operated at an absolutely uniform speed for the entire duration of the trial, the number of revolutions being such that the true speed of the vessel through the water, if there were no wind or current, would be exactly five knots per hour. If the first run is made over the mile course in the direction towards which the current is setting, and so that the time of the mid-instant of the run is exactly 1h. 00m. on the scale of time employed, it will readily be seen that the observed speed of the vessel over the ground will be 5.2 knots.

After completing this first run and returning over the measured mile in the opposite direction, the time consumed in the turn being such that the mid-instant of the second run occurs at 1h. 20m. the speed of the vessel over the ground will evidently be 4.6 knots. An average of these two observed speeds would give 4.9 knots as the corrected speed of the vessel, or 0.1 knot less than the true speed.

It will be noticed that if we had assumed that the current were uniformily decreasing (instead of increasing) with the elapsed time, the average speed of the two runs would have been 0.1 of a knot greater than the true speed instead of 0.1 of a knot less.

Consider now that the series of two runs which we have just described was prolonged by making a third run over the course under precisely the same conditions as were maintained for the first two runs, the mid-instant of this run occurring at 1h. 40m. of our scale of time (Fig. 6). The speed of the vessel over the ground in this case would be 5.6 knots and the average of the speeds for the second and third runs would be 5.1 knots.

It will therefore be seen, that when the current is ***increasing,***

if we start the first of a pair of runs with the tide we get an average speed which is slightly *less* than the true speed of the vessel; whereas, with the same current, if we start the first of a pair of runs *against* the tide, we get an average speed which is slightly *greater* than the true speed of the vessel.

It will further be noticed that with a current which increases or decreases at a uniform rate within the period of time considered, we obtain in either case the true speed of the vessel by making a series of three runs over the course, so spaced that the time intervals between the mid-instant of adjacent runs remains the same, and then taking the arithmetical mean of the first and second observed speeds, then that of the second and third, and finally a second mean of the first two means. In the case just cited our true speed is $\frac{1}{2}(4.9+5.1)=5$.

It is evident that we may obtain our true speed more directly than by the method of successive means, if we combine the three speeds observed on the course by the trapezoidal rule. Let s_1, s_2, and s_3, be the three observed speeds; the value S of the true speed is then given by the expression

$$S=\tfrac{1}{4}(s_1+2s_2+s_3),$$

or, substituting the observed values of the speeds in our numerical illustration given above, we shall have,

$$S=\tfrac{1}{4}(5.2+2\times 4.6+5.6)=\frac{20}{4}=5.$$

The point to which it is desired to call especial attention, is that when a vessel makes a pair of runs over a measured-mile course, under any condition of current other than that of a perfectly constant velocity, the value obtained for speed by taking the arithmetical mean of the apparent speeds as observed will be either in excess of or below the true speed.

It is important that this fact should not be lost sight of, as it is frequently the case that speed trials are conducted by making pairs of runs at as nearly as possible the same speed for each pair, and the curve of Speed Revolutions determined by drawing a curve through the average spots thus obtained. When it is remembered that to the irregularities in the ob-

served speeds, caused by the change in the velocity of the current, are to be added those due to the variations in the speed of the engine, it will be seen that the method of averaging a single pair of runs in order to determine a point on the speed curve is unsatisfactory and should never be employed unless reliable and independent data show that during the period of time considered the velocity of the current was constant, or the amount of its variation so small as to be properly neglected.

X. General Problem of Correction for Current.

Consider now the general problem of accurately calculating, from the data obtained on runs over a measured-mile course, a point upon the empirical curve expressing the relation between the true speed of a vessel through the water and the number of revolutions per minute of her propeller. The conditions under which the data are taken on the mile course, to be such as usually occur in practice—*i. e.*, the strength of the current upon the trial course is not constant and its amount is not known.

In reviewing the literature bearing upon this subject, its treatment by Naval Constructors Pollard and Dudebout, of the French Navy, is so satisfactory that I consider no apology necessary in laying before the Society a somewhat free translation of their remarks,* in the belief that they deserve to be more generally known.

"Naval Constructors Doyére and Bonvalet of the French Navy are the authors of a very interesting investigation† upon the best method of combining the results of a number of successive runs made in alternate directions over a measured course, upon which there are variable currents, in order to calculate in the most accurate manner possible the true speed of a vessel through the water, S, from the apparent speeds as measured on the runs over the course, $s_2, s_2, \ldots sn$.

*Pollard et Dudebout, Ingenieurs de la Marine. "Theorie du Naviere," Vol. IV, p. 303 (Paris, 1894).

†Doyere et Bonvalet, Ingenieurs de la Marine, "Note relative aux conditions des essais de recetti du Tage," (Brest, 23 Mai, 1889).

Suppose first that the sum of the components in the direction of the length of the trial course, of all those forces which produce a variation in the true speed of the vessel, are represented by a certain function of time, $f(t)$, the resulting variation in speed being either positive or negative according to the direction in which the vessel runs over the course. For a given run over the mile course, let

t_1 = the time of crossing the first range.
t_1 = the time of crossing the second range.

Then we have for the value of the apparent speed s,

$$s_1 = S \pm \frac{\int_{t_1}^{t'_1} f(t)\,dt}{t'_1 - t_1}. \quad . \quad . \quad . \quad . \quad . \quad (1)$$

It will be noticed that the second term of the right-hand member of this equation represents the influence upon the speed of the vessel, not only of the current, but also of the wind, since $f(t)$ is defined as the sum of the longitudinal components of all forces affecting the speed.

If in Fig. 7, Plate VII, we represent by a curve the value of $f(t)$ as a function of time, the second term of the right-hand member of equation (1) is the mean ordinate of the area $t_1\,t'_1\,BA$ and is approximately equal (with an error of the same order that is usually committed when we apply the trapezoidal rule to determine the area bounded by a plane curve) to the ordinate which corresponds to the instant $T_1 = \frac{t_1 + t'_1}{2}$, in other words to $f(T_1)$, so that the true speed of the ship, S, becomes

$$S = s_1 \pm f(T_1),$$

Under the above conditions, if we make a series of n runs alternately in one direction and then in the other over a measured-mile course (the first run being supposed in the direction of the current) and if we admit that the runs are sufficiently regularly spaced, so that the interval between the middle instant of successive runs is practically constant, we shall have

$$S = s_1 - f(T_1),$$
$$S = s_2 + f(T_1 + a),$$
$$S = s_3 - f(T_1 + 2a),$$
$$\cdots\cdots\cdots\cdots$$
$$S = s_{n-1} + (-1)^{n-1} f[T_1 + (n-2)a],$$
$$S = s_n + (-1)^n f[T_1 + (n-1)a],$$

and developing these relations in terms of the successive derivatives of the function $f(t)$, (which is always possible if we admit that the function $f^p(t)$ is continuous between the limits of time considered and that θ is a quantity comprized between zero and unity), we have—

$$S = s_1 - f(T_1),$$

$$S = s_2 + f(T_1 + af'(T_1) + \frac{a^2}{1\cdot 2} f''(T_1) + \ldots + \frac{a^{p-1}}{1\cdot 2 \ldots (p-1)} f^{p-1}(T_1) + \frac{a^p}{1\cdot 2 \ldots p} f^p(T_1 + \theta a),$$

$$S = s_3 - f(T_1) - 2af'(T_1) - \frac{(2a)^2}{1\cdot 2} f''(T_1) - \ldots \frac{(2a)^{p-1}}{1\cdot 2 \ldots (p-1)} f^{p-1}(T_1) - \frac{(2a)^p}{1\cdot 2 \ldots p} f^p(T_1 + \theta 2a),$$

$$\cdots\cdots\cdots\cdots$$

$$S = s_n + (-1)^n \left[f(T_1) + (n-1) af'(T_1) + \frac{(n-1)^2 a^2}{1\cdot 2} f''(T_1) + \ldots + \frac{(n-1)^{p-1} a^{p-1}}{1\cdot 2 \ldots (p-1)} f^{p-1}(T_1) + \frac{(n-1)^p a^p}{1\cdot 2 \ldots p} f^p(T_1 + \theta(n-1)a) \right].$$

Multiply both members of each of these equations except the first by the indeterminate numerical factors $\lambda_2 \lambda_3 \ldots \lambda_n$ and add, we shall have

$$S(1 + \lambda_2 + \lambda_3 + \ldots \lambda_n) = s_1 + \lambda_2 s_2 + \lambda_3 s_3 + \ldots + \lambda_n s_n.$$
$$+ f(T_1)[-1 + \lambda_2 - \lambda_3 + \lambda_4 + \ldots],$$
$$+ af'(T_1)[\lambda_2 - 2\lambda_3 + 3\lambda_4 - 4\lambda_5 + \ldots],$$
$$+ \frac{a^2}{1\cdot 2} f''(T_1)[\lambda_2 - 2^2\lambda_3 + 3^2\lambda_4 - 4^2\lambda_5 + \ldots],$$
$$+ \frac{a^3}{1\cdot 2\cdot 3} f'''(T_1)[\lambda_2 - 2^3\lambda_3 + 3^3\lambda_4 - 4^3\lambda_5 + \ldots].$$
$$\cdots\cdots\cdots\cdots$$

The number of runs over the course being n we can select such values for the $(n-1)$ indeterminate factors $\lambda_2, \lambda_3 \ldots \lambda_n$ as to render simultaneously zero the coefficient of $f(T_1)$ and its $(n-2)$ first derivatives, which will give a system of $(n-1)$ equations of the first degree, containing $(n-1)$ unknown quantities.

These being solved and the indeterminate factors thus assigned numerical values, there will remain—

$$S(1+\lambda_2+\lambda_3+\ldots+\lambda_n)=s_1+\lambda_2 s_2+\lambda_3 s_3+\ldots +\lambda_n s_n+Af^{n-1}(T_1)+Bf(T_1)+\ldots \quad (1)$$

hence,

$$S=\frac{s_1+\lambda_2 s_2+\lambda_3 s_3+\ldots+\lambda_n s^n}{1+\lambda_2+\lambda_3+\ldots+\gamma_n}+\frac{Af^{n1}(T_1)+Bf_n(T_1)+\ldots}{1+\lambda_2+\lambda_3\ldots+\lambda_n}. \quad (2)$$

Now admit that the true speed S, has the value given in (3) instead of that given in (2)

$$S=\frac{s_1+\lambda_2 s_2+\lambda_3 s_3+\ldots+\lambda_n s_n}{1+\lambda_2+\lambda_3+\ldots+\lambda_n}. \quad (3)$$

which amounts to admitting that the derivatives of the function $f(t)$ of an order equal and superior to $(n-1)$ are simultaneously zero—that is to say, by replacing the curve representing the force of wind and current, Fig. 7, within the region corresponding to the total duration of the n runs, by a parabolic curve of the $(n-2)$ degree which can be made to pass through the $(n-1)$ points of the true curve, and which will approach the true curve with the greater exactitude for a given total duration of trial, according as the runs are the more numerous and the intervals of time between them the shorter.

If n is given values of the successive integers it follows that,

For $n=2$,

$\lambda_2=1$.

The current curve is here assumed to be a straight line parallel to the axis of abscissæ (this is the case of a constant current);

For $n=3$,

$\lambda_2=2, \lambda_3=1$.

The current curve is assumed to be a straight line, making any angle whatever with the axis (this is the case of a velocity of current increasing or decreasing in proportion to the elapsed time);

For $n = 4$,

$\lambda_2 = 3, \lambda_3 = 3, \lambda_4 = 1$.

The current curve is replaced by a parabola of the second degree having three points in common with the true curve;

For $n = 5$,

$\lambda_2 = 4, \lambda_3 = 6, \lambda_4 = 4, \lambda_5 = 1$.

The current curve is replaced by a cubic parabola having four points in common with the true curve;

For $n = n$.

The values of the coefficients $\lambda_2, \lambda_3 \ldots$ are the numerical coefficients in the development into a series, by the binominal theorem, of the function $(a+b)^{n-1}$.

It may be remarked that the method just described corresponds to the English method of continued averages, which consists in first taking the mean value of the apparent speeds two by two, then second means two by two of the first mean values, then third means two by two of the second means, and so on. This will be seen if in the above expression (3) for the true speed, S, we take $n = 5$ for example; then

$$S = \frac{s + 4s_2 + 6s_3 + 4s_4 + s_5}{16},$$

and with the English method we have,

$$s_1, \quad s_2, \quad s_3, \quad s_4, \quad s_5,$$

$$\frac{s_1 + s_2}{2}, \quad \frac{s_2 + s_3}{2}, \quad \frac{s_3 + s_4}{2}, \quad \frac{s_4 + s_5}{2},$$

$$\frac{s_1 + 2s_2 + s_3}{4}, \quad \frac{s_2 + 2s_3 + s_4}{4}, \quad \frac{s_3 + 2s_4 + s_5}{4},$$

$$\frac{s_1 + 3s_2 + 3s_3 + s_4}{8}, \quad \frac{s_2 + 3s_3 + 3s_4 + s_5}{8},$$

$$\frac{s_1 + 4s_2 + 6s_3 + 4s_4 + s_5}{16} = S.$$

In accordance with what has been said, in applying this method to runs over a measured course, the number of runs n should be taken proportionately greater, according as the true speed of the ship, S, is itself the higher, and the endeavor should be made to limit the time for making the turns to that strictly necessary. If there is a choice between two measured courses, it is desirable to use the shorter unless on account of the shortness of the course there should not be sufficient time to accurately measure the indicated horsepower, or that the errors due to marking the time when crossing the ranges should not be considered negligible, under which circumstances the longer course should be used. Finally, the time intervals between the middle of the runs should be spaced as regularly as possible.

On the other hand, the total time comprised in the runs (including the time necessary to turn while off the course) may reach four or even six hours. Under these conditions it should be remarked that, if the increase in the number of runs increases the accuracy from the point of view of elimination of the effect of current, on the other hand the prolongation of the length of the trial increases the error committed on account of other disturbing causes which obey laws altogether different from that due to the current; such, for example, as the choking of the fires towards the end of a prolonged trial, which results in a gradual decrease in the steam pressure, consequently, in the number of the revolutions of the screw and the corresponding true speed of the ship through the water. Under these circumstances the speed, instead of being constant, is then a function of time $S=\varphi(t)$ and, for this new variation in the speed, there is but one method of elimination which consists in taking a simple arithmetical mean of the observed speeds.

In consequence Messrs. Doyére and Bonvalet advise the adoption of an intermediate method, which consists, no matter what the number of runs may be, in grouping them four by four in the following manner:—

$$s_1,\ s_2,\ s_3,\ s_4,$$
$$s_2,\ s_3,\ s_4,\ s_5,$$
$$s_3,\ s_4,\ s_5,\ s_6,\ .$$

and so on. That being done, the speed is calculated for each group by tripling the two intermediate speeds, adding the end speeds and dividing by eight in each group, *i. e.*,

$$\frac{s_1 + 3s_2 + 3s_3 + s_4}{8}$$

The values thus obtained permit the curve $S = \varphi(t)$ to be traced. Then the arithmetical mean is taken of the values thus obtained, and this mean value gives the true speed of the vessel, S, corresponding to an average condition of the fires. It will be seen that for six runs this intermediate method gives for the value of the true speed—

$$S = \frac{s_1 + 4s_2 + 7s_3 + 7s_4 + 4s_5 + s_6}{24},$$

whereas, the original method would give—

$$S = \frac{s_1 + 5s_2, + 10s_3 + 10s_4 + 5s_5 + s_6}{32}.$$

As a result of the investigation of Messrs. Doyére and Bonvalet and that of a commission of the French Admiralty, M. de Maupeou, chairman, the following directions were adopted in the French naval service from the date of July 31, 1890:—

I. The arithmetical mean of the speeds determined by an even number of alternating runs over a measured course will be taken—

(1) Where no tide exists, or where the tide is of little importance.

(2) Where a tidal current exists, but when an ample number of runs can be made, and under suitable conditions, so that the errors resulting from the variations in the current are practically mutually compensated for.

II. On trial courses where a tidal current does exist and

when only a limited number of runs can be made, or when the runs are made under conditions which do not permit the elimination of the effect of the variations in the current by employing the preceding method, the runs will be grouped three by three, the mean of each group being obtained by doubling the middle one, adding the first and third and dividing by four; finally, the arithmetical mean will be taken of the results of each group, thus obtained.

The grouping of the runs three by three may be performed in several ways, but it is important that too great a weight should not be given to the result obtained on any one run, and for that reason a greater weight than two is not assigned to any one observation, which leads to the following grouping:—

For three runs, one group,

$$\frac{s_1 + 2s_2 + s_3}{4} = S.$$

For five runs, two groups,

$$\frac{s_1 + 2s_2 + s_3}{4}; \quad \frac{s_3 + 2s_4 + s_5}{4}.$$

Mean,

$$\frac{s_1 + 2s_2 + 2s_3 + 2s_4 + 2s_5}{8} = S.$$

For six runs, two groups,

$$\frac{s_1 + 2s_2 + s_3}{4}, \quad \frac{s_4 + 2s_5 + s_6}{4}.$$

Mean,

$$\frac{s_1 + 2s_2 + s_3 + s_4 + 2s_5 + s_6}{8} = S.$$

For seven runs, three groups,

$$\frac{s_1 + 2s_2 + s_3}{4}, \quad \frac{s_3 + 2s_4 + s_5}{4}, \quad \frac{s_5 + 2s_6 + s_7}{4}.$$

Mean,

$$\frac{s_1 + 2s_2 + 2s_3 + 2s_4 + 2s_5 + 2s_6 + s_7}{12} = S."$$

XI. Application of the Preceding Investigation to the Determination of a Curve of "Speed—R.P.M."

It will be seen that the above methods apply entirely to the determination of a single true speed, all the runs being presumed to be made with as nearly as possible the same number of revolutions of the engine. Further, it is assumed that the runs in question are made at the maximum speed that the vessel is capable of developing as otherwise the considerations advanced as to the decrease in speed due to the fall in steam pressure, owing to the choking of the fires by ashes, etc., at the end of a prolonged trial, would not apply.

The fulfillment of trial requirements which only necessitate the determination of the speed that the vessel can make at her maximum power while running over a measured-mile course, permits a simpler method to be adopted for the performance of the trial than is necessary when the contract requirements are of a more searching character. In the case of vessels built for the United States Navy, the contract speed is usually required to be maintained during a four-hours' run at sea; and the preliminary runs over the measured-mile course must give not only the maximum true speed developed on the mile, but the true speeds obtained at certain reduced powers, so that sufficient data may be available to determine a part of the curve of true speeds and R.P.M. of the engine. By means of this curve we determine just what speed has been developed during the four-hour trial, and whether the contract requirement as to speed has been complied with; or, in case the contract speed has not been attained, what is the exact amount of the deficiency. This determination must be sufficiently accurate to permit the application of a penalty that in certain cases may reach the rate of $400,000 per knot.

Fortunately, with the proportion of boiler power allowed vessels of the United States Navy of recent design the difficulty of maintaining speed during the four-hour trial on a straight away course is not much greater, if indeed it is as great, as the maintenance of that speed for five or six runs

over the measured mile. For example, in the case of the cruiser (*Charleston*) the highest speed developed on the standardization trial was 21.95 knots as the average of two runs, whereas on the four-hour trial a speed of 22.04 knots was maintained.

The speed on a successful four-hour trial is, therefore, apt to be very close to that of the series of runs made at the highest speed over the measured-mile course. Consequently, if we neglect for the moment all consideration of the information the speed and power curves may give bearing on the general performance of the machinery, propeller, etc., and limit the number of runs to that strictly necessary to obtain a reliable result in so far as the speed maintained during a four-hour trial is concerned, it may be said that the absolute minimum of information required, consists in—

1. The *exact* determination of the true speed through the water and the corresponding number of revolutions of the engines when the vessel is run at her highest speed over the mile course.

2. The accurate determination of the *direction* of the curve of "Speed—R.P.M." where it passes through the point just established.

In considering the first of these requirements, especial weight should be given to the fact that, even with a properly established trial course, our observations are subjected to accidental errors owing to irregular fluctuations in the current and to variations in the personal equations of the observers, so that a considerable number of observations are essential if reliance is to be placed on the result. The minimum number of runs that would appear to be satisfactory for this purpose is five. The French Admiralty method of combining the results of this number of runs, as previously described (viz—first obtaining an average speed for a sub-group comprising the first, second and third runs, using in the calculation of this average the formula $\frac{s_1 + 2s_2 + s_3}{4}$; then obtaining a similar average speed for the sub-group composed of the third, fourth and fifth runs; and finally taking the arithmetical mean of the

two average speeds so determined) is especially desirable since it permits an immediate appreciation *during the trial* of the regularity with which the current is varying. This follows from the fact if the speeds resulting from the averages of these two sub-groups differ to any noticeable extent, it may be immediately observed and the number of runs at this highest speed may be increased until a sufficient number are obtained to insure that irregularities in the results due to errors in one or more of the observations shall not materially affect the final value determined for this highest speed.

The determination of the direction of the curve at its upper extremity theoretically requires only a knowledge of the position of one other point in the curve, which should be located sufficiently near the point of highest speed to permit a line joining the two points to represent the tangent to the curve at its highest point. With this information, the speed on a four-hour trial could be accurately determined whenever, as in the case of the *Charleston,* this speed only differed from the highest speed observed in the standardization trial, by a small fraction of a knot. The accuracy of this method of the determination of the speed on a four hours' trial, would however rapidly decrease as the difference between the speeds increased; and such a method would, moreover, afford the minimum general information as to the efficiency of the ship and its machinery, for a given number of runs over the measured mile. This is evident, since to construct the tangent we must know the position of the second spot on the curve with the same accuracy as the first, which would require five additional runs at a speed only slightly lower than the highest speed obtained on the mile course. Ten runs would, therefore, have been made, all at powers requiring forced draft to the maximum, or nearly maximum permissible amount, and the net result would be the determination of two neighboring points on the curve of "Speed—R.P.M."

If instead of limiting the investigation of the speeds developed on the measured mile, so that it will furnish only the minimum amount of information required to determine the

speed during a four-hour full-power trial, a larger number of points of the curve of "Speed—R.P.M.," is established through such a range of speeds that a considerable part of the upper portion of this curve is determined, it will be found that although the number of runs over the mile course is increased, the difficulty of conducting a successful trial is diminished, since the number of runs at the higher speeds may be materially reduced. The propriety of this reduction will be plainly seen if it is remembered that since it is now proposed to determine a number of points on this curve, the runs made at other than the very highest speed can be made in a series of gradually reduced speeds, for which the corresponding horsepowers fall even more rapidly. The desirability of making a number of runs at considerably less than full power is the more evident if it is further remembered that since we are now determining the direction of the curve at its upper extremity by establishing the position of a number of its points, it is obvious that if we restrict the observations made to the immediate neighborhood of the upper extremity of the curve, irregularities in the observations of any given amounts will affect the resulting direction of the curve more than if they had occurred in the determination of points further removed from the curve's upper extremity.

In conducting a standardization trial in a rational manner, there are, therefore, two separate phases—1st, the *exact* determination of the point on the curve of "Speeds—R.P.M.," corresponding to the highest speed over the mile course; 2d, the approximate determination of a number of lower points on this curve, by means of which the upper portion of the curve may be drawn, so that the *direction* of the curve at its upper extremity may be accurately known.

The establishment of the position of these lower points of the curve, may be accomplished by either one of two methods. The first consists in determining each of them separately, by a series of runs each entirely independent of those which precede or follow it, and therefore not liable to be seriously interfered with by any interruption of the trial that might occur

from any unforeseen cause. The series of runs to determine each of these lower points of the curve should consist in not fewer than three runs for each of them, as is obvious from what has been previously said relative to determining the true speed from observations made of the speeds with and against a variable current. Four is the minimum number of points of the curve which should be established in addition to the point for the highest speed, this number being selected in belief that in physical measurements of this character, no empirical curve should be determined by fewer than five points. It is believed to be manifest that in an investigation of this character, it should be possible to so group the data collected, that any accidental irregularity in certain of the observations, might be detected by plotting the results and running a fair curve through the points so obtained; any point falling off this curve would be neglected as of doubtful accuracy. With one point so neglected, the remaining four would still permit the curve to be constructed, but with fewer than four points the accuracy of the results would be too much open to question to permit them to be used in applying a heavy money penalty for a deficiency in speed below contract requirements.

The second method of establishing the lower portion of the curve of "Speed—R.P.M.," is one first suggested by Naval Constructor D. W. Taylor, U. S. N.* This method consists in making a continuous series of runs with a constant increase or decrease in the number of revolutions made on each run, and with the time interval between the mid-instant of each run and the run adjacent to it maintained as nearly constant as possible. The runs are made in alternate directions with and against the tide, only one run being made for each speed of engine, and the results are plotted as two separate curves of speeds and revolutions, one with and one against the current. The curve of true speeds is then obtained by taking from the two curves thus established the mean values of the speeds with and against the tide at constant numbers of revolutions.

*"JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS," page 587, Vol. IV, 1890.

This method will probably permit equal accuracy to be obtained with a less number of runs than in the case of the first method, provided the vessel tried has a well trained crew so that the somewhat onerous conditions as to the regularity of the intervals of time and of revolutions are maintained; and moreover that the trial is not interrupted by vessels crossing the trial course at such a time as to interrupt the series of runs and cause the trial to be repeated.

This second method is, therefore, well adapted for experimental investigations of the qualities of a naval vessel in commission, over a trial course where there is little probability of interruption by passing vessels.

To compare the number of runs required with these two methods in order to establish five points on the curve of true "Speed—R.P.M." with the same probable degree of accuracy, it will be necessary to have—

With the first method—five runs at the highest speed and four series of three runs each, at the lower speeds, or seventeen runs in all (Fig. 8, Plate VII).

With the second method—five runs at the highest speed and nine runs at lower speeds, or fourteen runs in all (Fig. 9, Plate VII).

The number of points selected to determine each of these curves is five, and although each of the points in the curve of true "Speed—R.P.M." as determined by the first method is the result of three runs, so that accidental error is in part eliminated by the greater number of observations; whereas in the second method, while the two curves of runs with and against the tide are each plotted by five points based upon a single run each, yet the fact of the runs forming a series, permits a graphic correction for accidental errors that is probably quite as effective as the greater number of runs in the first method, in securing accuracy in the final result.

It should be noted, however, that with the second method in case of an interruption lasting such a length of time that the curves of "Speed—R.P.M." with and against the tide, after the interruption can no longer be considered as a prolongation

of the similar curves made before the interruption, it is necessary either to repeat the whole series of runs, or such a part of it as will give such an amount of overlap between the two parts of each of the curves of runs with and against the tide, as will permit each of these curves to be satisfactorily established. In case of interruption, the second method may, therefore, require more runs than is necessary with the first method in order to obtain equal accuracy.

For these reasons it appears to be preferable in a contract speed trial for the lower portion of the speed curve to be determined by separate points, each determined from the results of three runs over the mile course at the same revolutions of the engines. In choosing these points, they should be so selected as to accurately define the direction of the upper part of the "Speed—R.P.M." curve; the lowest point on this curve should correspond to about one-half full power, and between that speed and the highest speed three intermediate points should be established.

As to the order to be followed in making the runs it is believed that in any case, no matter which method of trial is used, the high-speed run should be made first, for with large vessels the time required to make these five runs should never exceed two and one-half hours, unless the trial is interrupted, and the succeeding runs being made at reduced powers would at once relieve the strain on the engineering force of the contractor conducting the trial, a matter of no small importance where the number of highly trained men in charge of the machinery is naturally limited.

It is interesting to note, in connection with the consideration of the minimum number of runs required to properly conduct a standardization trial for the purpose of determining the speed of a vessel on her contract trial, the various opinions that have been expressed.

Thus Rear Admiral Melville, U. S. N., writing on January 13, 1890, relative to the proper conditions of speed trial of 4,520-ton cruisers of 19 to 20-knot speed, recommended that runs be made at four different speeds, 14, 16, 18 knots and the

highest speed practicable, and that six runs be made at the highest speed and four runs at each of the three lower speeds, or 18 runs in all. Chief Engineer Isherwood,* U. S. N., recommended where extreme accuracy was desired, not less than ten double runs for each intended speed.

The Navy Department in its instructions† to the Trial Board of the U. S. S. *Bancroft,* a gunboat having a contract speed of 12 knots, directed that one pair of runs be made at 7 knots, one pair at 9 knots, two pair at 11 knots, two pair at 12 knots and two pair at a higher speed—or 16 single runs in all. The Board actually made 20 single runs in the trial of the vessel.

It will be seen that the number of runs above recommended, *i. e.,* 17 single runs for the first method, or 14 single runs in case an experienced crew permits the second method to be used, is below rather than above the earlier requirements. The necessity for at least the number of runs recommended will be apparent if the current curves in Plate X are examined carefully, and it is noted how far the assumption of uniformly increasing or decreasing currents is removed from the actual conditions.

XII. Practical Results Obtained With the Standardization Method.

Although the method of determining the speed developed by a vessel during a four-hour contract trial at sea, by means of a previous "standardization of the screw" has been used in conducting the trials of a large number of vessels built for the United States Navy, yet, with the exception of the recent trials of the cruiser *Charleston* and the battleship *Rhode Island,* these vessels have been torpedo boats and destroyers or small gunboats and cruisers of moderate speeds.

In the case of the torpedo boats and torpedo-boat destroyers, the circumstances of their trials, in so far as concerns cor-

*"Speed Trials of Fast Ships," Journal of American Society of Naval Engineers, Vol. II, p. 95 (Feb. 1890).

† "The contract trial of the U. S. S. *Bancroft*" by R. S. Griffin, Passed Assistant Engineer, U. S. N. Journal of Am. Soc. Naval Engineers, Vol. V, p. 377 (May 1893).

recting for *variations* in the currents encountered on the mile course, was a very different matter from that necessary when high-speed vessels of large displacement are tried. Owing to the very short interval of time required by a torpedo boat or destroyer to acquire its full speed due to a given horsepower, it is possible to make quick turns at each end of the mile course, with the result that the total time required for a destroyer to make a series of runs over the course will be only a fraction of the time required by a larger vessel.

The short interval of time between successive runs with a torpedo boat permits, therefore, the assumption without serious error that a pair of runs are made under practically constant strength of current.

In the trial of large high-speed vessels, such as modern battleships and armored cruisers, the time required to make a complete series of runs of sufficient number for reliance to be placed upon the accuracy of the results, may be taken as extending from four to six hours, with intervals between the runs averaging between fifteen and twenty-five minutes. In such cases it is manifestly impossible to neglect the change in current which has occurred during the interval between successive runs.

It may be further remarked that if we consider the character of the curves of "Speed—I.H.P." of the smaller vessels, we will find in most cases that at their contract speeds the ratio of the increase in horsepower necessary for small increase in speed is moderate in amount, while with the large high-speed vessels this ratio is usually much higher. In other words, in the first case the vessel readily responds to the demand for a slight increase in speed; while in the second case the conditions may be such that a slightly higher speed than that called for by the contract is extremely difficult of attainment, and it is, therefore, necessary that every reasonable precaution be taken in the case of these large high-speed vessels to accurately determine exactly what speed is maintained during trial.

As an illustration of the observations taken on a standardization trial, there is shown on Plate IX the results of a stand-

ardization trial of the armored cruiser *Colorado,* made on the measured-mile course at Provincetown, Mass., on September 14, 1905.

It will be seen that this trial consisted in making a series of runs over the mile course at constantly increasing numbers of revolutions per minute of the engines for each succeeding run, the attempt being made to have the revolutions increased by regular increments until the highest safe engine speed was attained, when three runs were made over the course at that engine speed. Each run was made over the course in the opposite direction to that of the preceding one, and the time intervals between the mid-instant of successive runs was maintained as uniform as possible. The data, from which the curve of "Speed—R.P.M." shown upon the upper portion of this sheet, was plotted, are given in the first table of Plate XII.

Unfortunately the conditions on this trial for obtaining data as to indicated horsepower were not satisfactory, and the horsepower used in plotting the curve of "Speed—I.H.P." are only roughly approximate, as they are based upon the power developed by the starboard engine alone (the indicators on the port engine not working in a satisfactory manner). Other unsatisfactory conditions as to the determination of the I.H.P. existed, as only one set of cards were taken during each run over the course and the indicators used were not of the "exposed spring" type, but had the spring enclosed so as to be subject not only to the heat from the steam but also to the radiation from the walls of the indicator body.

The first point in connection with the curves shown on Plate IX to which it is desired to call attention, is the divergence between the curves of "I.H.P." and "R.P.M." for the starboard engine, when plotted separately according to the direction in which the vessel was running over the course. Thus at 90 revolutions per minute, while going south, the horsepower developed by this engine was about 4,700, whereas, when going north, the I.H.P. required for the same number of revolutions of the engine was about 5,250.

The divergence between these two curves representing the results on the runs to the north and to the south decreases gradually towards the end of the trial. During the trial the wind decreased from a force of 4 (by Beaufort scale) at the beginning, to a force of 3 at the end, while at the same time its direction changed so as to be less ahead. It appears probable that (neglecting a part of the difference between these two curves of horsepower, as being due to the unsatisfactory conditions under which the power was determined) the effect of the wind upon the vessel is the cause of the divergence noted between these two curves.

To make an approximate estimate of the effect of this wind upon the speed of the vessel take, for a given number of revolutions, the difference in power between the point on the curve representing the power developed during the run against the wind (curve "Going North") and that for the run before the wind (curve "Going South"). This amount of power (which is for the starboard engine alone) is doubled to give the approximate difference in power for both engines, and then this increment of power is applied to the curve of "Speeds—I.H.P." from which it will be seen that for the point just considered (90 revolutions) the corresponding difference in speed for the same I.H.P., according as the run is made before the wind or against it, is about 0.6 of a knot. This amount is undoubtedly excessive, owing to the very unsatisfactory determination of the data from which these curves of power were constructed, but it is sufficiently large to call attention to the important character of the effect of the wind in the determination of the speed of large, fast vessels.

It should be noted that since the wind pressure is proportional to the square of its velocity, the mean curve, drawn half way between the curves showing the results of runs before and against the wind, does not represent a curve showing the results of runs when there is no wind. And the same fact may be put in another form, by stating that, when making runs with and against the wind over a measured-mile course, the wind will not help the vessel when steaming before it as

much as it retards when steaming against it; as a consequence the standardization trials should not be made when there is a strong wind, as such conditions are always unfavorable to the vessel tried.

Returning now to a consideration of the other curves shown on Plate IX, it will be seen that the curves giving the "Speeds and R. P. M." for runs with and against the tide permit the construction of curves showing the effect of current and wind upon the speed of the vessel at the time each run was made.

At the time the *Colorado* was standardized, a gunboat, anchored half-way between the two ranges, and about 200 yards from the line of the course, took observations of the direction and amount of current with a current pole, as has been done on the trials over the Cape Ann Course. The results of these observations are given in curve "C" at the bottom of Plate IX, together with the direction and force of wind (the latter by Beaufort scale).

Finally, a comparison is made of the curve (curve "B") of the components in the direction of the course of the current velocities, as measured with a current pole, and the corresponding curve (curve "A") of the components of the effect of both wind and current in the direction of the course. Curves giving similar comparisons for several other standardization trials are shown in Plate X. Such a comparison is interesting as showing what grossly inaccurate results the application of the velocities observed by the current pole would give, if applied to the observed speeds over the mile course. And here the vessel taking the current observations is never more than half a mile from the vessel running over the course, whereas, on the Cape Ann Course, this distance may exceed three miles. It will be noticed that the only time there is, in general, any agreement between the two curves is at slack water, when there is no current. At other times the difference will frequently reach half a knot.

In Plate XI and XII are shown type forms prepared to record graphically and in tabular form in as condensed a manner as possible the results of a standardization trial and the

more important coefficients of performance, and a comparison with the model tank curve of effective horsepower. The results of the vessel's performance during the four-hour speed trials, both preliminary and final, and general data as to her dimensions and general characteristics are also given. It is especially desirable that the engineering data taken during standardization trials should be complete, and it is regretted that in the case of the type form, which has been taken as an illustration, there should be such a paucity of information available.

There is one point in connection with the engineering data of such a trial that deserves especial mention. It has been seen that one of the assumptions upon which the accuracy of either of the methods used for conducting a standardization trial is based, is the ability to operate the engines at predetermined speeds. Now, with a new vessel and an untrained crew this is, at the best, by no means readily done. It will, however, be greatly facilitated if, previous to the standardization trial, the vessel is run at four or five different speeds, covering as great a range of engine speeds as possible; and, while both engines are running steadily together at each of these speeds, the revolutions per minute and corresponding steam pressure at the H.P. valve chest are noted for one engine. These results being put in a curve, the steam pressure necessary to produce any intermediate number of revolutions is readily interpolated. It is, of course, necessary that the amount of cut-off in the different cylinders should not be changed during this time; or if it is changed that separate curves be made for each condition of cut-off—as, for example, are shown by Mr. Charles P. Wetherbee for the actual results of the trial of the torpedo boat *Biddle* in Plate 48, accompanying his paper on "Changes in Torpedo-Boat Design," read in 1901 before the Society. I take pleasure in acknowledging my indebtedness to Mr. Wetherbee in connection with the preparation of the table for engineering data contained in Plate XII, for advice which his experience with the standardization trials of high-speed vessels renders especially valuable.

Having selected one engine as the pacemaker during the standardization trials, the other engine is run in unison with it. It is very desirable that there should be supplied for such trials a differently geared pointer which will show immediately and on a large scale, any change in the relative speeds of the two main engines.

In Plate XIII is shown the relation between the speeds of the port and starboard engines of the *Colorado,* during her standardization trial at Provincetown on September 14, 1905. The starboard engine was here taken as a pacemaker and, in general, it was run very steadily while on the course. The port engine, on the contrary, ran much more irregularly, and its record shows plainly the desirability of better arrangements for maintaining uniformity in speeds of the two engines.

For it should be noted that in all that has preceded as to the general discussion of the standardization of the screw method of speed trial, the speed of a single engine has been referred to. With two or more screws it is necessary to take the average revolutions of all the screw shafts, and it is desirable that the difference between the revolutions per minute of the main engines on each run over the mile course should be as small as possible.

It may be of interest to describe briefly the method by which the revolutions of the engines are recorded while the vessel makes a run over the measured-mile course.

This consists in two observers, one stationed on the forward bridge and one on the after bridge, each closing an electric circuit at the instant that they pass across the ranges which mark the mile course. The act of closing each circuit causes a printing device in each shaft alley to record upon a paper tape the position of five (5) revolving dials which were driven by suitable sprocket wheels and chain from the main engine shaft. Each of the five (5) dial wheels was geared to a ratio of one to ten to the wheel next on its left, and the printed tape showed the position of the engine shaft to within one-tenth of a revolution.

The instruments used in the shaft alleys are of the type used by Naval Constructor Taylor in making records of the speeds of fans and are both accurate and reliable when properly installed. They have, however, elaborate mechanical feeds for the paper tape upon which the record is printed, which are not suitable for ship use, and a simpler instrument, giving a record of the same character, can be designed that will be more suitable for trial purposes on board ship.

The time required for the vessel to go over the mile course was obtained from the readings of stop-watches which were started and stopped by a suitable device contained in the circuit maker which actuated the printing mechanism in the shaft alleys. By this means any error in time, due to the personal equation of the observer, makes a corresponding error in the total revolutions on the mile, so that the personal error of the observer is practically eliminated from the computed R.P.M. of the engines. A photograph of this device for operating the stop-watch and circuit maker simultaneously, is shown in Plate XVI.

A third stop-watch, operated by an independent observer, furnished an additional time record, supplementing the times obtained by the two observers on the forward and after bridges. In averaging these observed times, any observation that differed by more than three-tenths of a second from the mean value of the other two observations was discarded.

With improved beacons, of the type of the proposed permanent beacon shown in Plate XV, properly located so as to give the best results, it is believed that the accuracy of observation can be increased sufficiently to fully justify the use of a chronograph to record the speed of the vessel instead of the stop-watches as described. It appears to be entirely possible with the stop-watch to have errors in the elapsed time of .4 of a second entirely due to the necessary conditions of operating the second hand. At high speeds the advantages attending the use of a chronograph appear to entirely justify the greater complication attending its use.

XIII. Conclusions.

I. In view of the present state of the art of construction of vessels for military purposes, it is essential to insure efficiency that the contracts for such vessels should provide, not only for a full-power trial of a few hours' duration, but also for additional trials each lasting at least 30 hours in order to determine—

(*a*) The endurance of the propelling machinery at the highest power that can be *continuously* maintained at sea.

(*b*) The economy of coal consumption at a cruising speed—approximately that corresponding to the maximum cruising radius of the vessel.

It is believed to be imperative, in order to properly care for the materiel of the vessel, that such trials should be conducted by a naval crew, the builder to have, however, as many men on board as he may desire and to retain full control and responsibility for the operation of the machinery which he has built. Such a procedure would follow the practice of all other first-class naval powers.

Exception to the above method of conducting contract trials should only be made in the case of torpedo boats, destroyers, etc., where, for reasons which are sufficiently obvious to anyone who has been concerned in the design or construction of such vessels, it is undesirable to introduce any division of responsibility that can by any possibility be avoided.

II. In determining the speed of a vessel during any of the contract trials referred to above, the double run over a long measured course at sea can only give reliable results in a tideless sea. When tidal currents do exist, the errors introduced by the attempt to measure the current are of the same order of importance as the effect of the current itself.

It is difficult to assign a definite value to either the probable or the maximum error that is introduced into the corrected speed, when a trial is conducted under these circumstances. The records you have before you speak for themselves. Personally, I believe that a current correction obtained by the

methods recently in vogue may introduce an error into the corrected speed making it as much as ¼ knot greater or less than the true speed.

To treat a serious matter lightly, I cannot but express my sympathy with an officer of one of our largest shipbuilding companies, who said:—"After we have run our trial over the Cape Ann Course and get our observed speed, the tidal correction is determined by drawing a number out of a hat."

III. In the standardization of the screw method of determining speeds at sea, we have the most accurate instrument of investigation that is now available for this purpose. Its results, however, like all other physical measurements, are only accurate to a certain degree, and this degree depends upon several conditions, amongst which the most important are—

First. Selecting a suitable trial course upon which to conduct the standardization trial.

Second. Marking the trial course properly, and perfecting each element by which the revolutions of the screw and the apparent speeds over the bottom are determined while the vessel runs over the course.

Third. Repeating the number of the observations, so that by averaging several similar results the effect of accidental error may be eliminated to as great a degree as possible.

Let us consider these conditions in the order just given.

First. The trial course should be sheltered so as to have relatively smooth water. It should have an adequate depth of water on it, and on the approaches to it. The general character of the tidal currents existing on the course should be carefully investigated so that it may be possible to estimate before the trial is held the times when the currents may be expected to be regular in character, so that periods of probably irregular fluctuations in the strength of the current may be avoided.

It should be noted that the absolute amount of the current is not material, but the more it departs from uniformity in the rate at which it increases or decreases, the greater will be the probable error that may be introduced into the corrected speed.

Second. The most important items under this head relate to the proper design and location of the beacons marking the ranges at the ends of the mile course.

(*a*) The beacons should have a base of ample size so that they can be immediately located from the ship by the unaided eye.

(*b*) This base should have a rounded, conical, or pyramidal form, so that it may be well lighted under all angles of azimuth of the sun, and its color should be distinctive.

(*c*) The large base of the beacon should be surmounted with a cylindrical pole, centered over the base, and no larger in diameter than is necessary for it to be seen from the ship. Each pole should be surmounted by a small pyramidal shape to assist in distinguishing it at a distance. The images of the poles of the front and back beacons should transmit *across* each other, as viewed from the ship, when crossing the range. For one pole to be entirely above the other, when viewed from the course, will not give satisfactory results.

(*d*) It is important, if the course is some distance from the beacons, that the poles should show against the sky as a background, so that its greater luminosity will render the slender poles distinctly visible. If this is manifestly impracticable, it has been suggested that an artificial background, that will remain the same color in either winter or summer, would materially aid in obtaining accuracy of observation.

(*f*) On any one course, the beacons on each range should be so located that the distance from the ship to the front beacon bears a ratio to the distance between the two beacons that is the same for each of the two ranges. The poles on each of the front beacons should have the same diameter and the back beacons should also have poles of the same diameter.

Passing now to a consideration of other items under this head, the present arrangement for recording the revolutions of the engine are admirable in general principle, but the construction of the shaft-alley machines should be simplified, the time feeds of the paper tape should be done away with, and

they should be so geared as to give the revolutions to hundredths.

When proper beacons are provided to mark the ranges, the stop-watch will be found not to be a sufficiently accurate instrument for marking the times of crossing the ranges when vessels of high speed are standardized, and a chronograph, reading to hundredths of seconds, should be used to record the times.

The greatest error to be anticipated under this general head, provided proper precautions have been taken to insure accuracy in observations, is a variation in the personal errors of the observers marking the times. For this reason all times should be taken by not less than three observers, and when any observer differs materially from the mean of the other two, his time should be discarded.

Third. In a contract trial, the vital point on the curve of "Speed—R.P.M." is the point determined nearest the contract speed. This point should be determined, for the reasons previously explained, by not less than five runs, arranged in two groups of three runs each. Where the speeds given by the two groups, *compared at the same number of revolutions per minute of the engines,* show a difference of speeds for the same revolutions, exceeding a slight amount (say .05 knot), the series of runs should be extended by additional runs until such a condition does maintain.

Having accurately established this upper point of the curve, the general direction of the curve is best established by determining a number of points on the curve situated at some distance from each other, so that slight errors in their position would affect as little as possible the *direction* of the curve where it passes through the point of highest speed just established.

For a contract trial, where it is desired to limit the number of runs as much as possible, four additional points of the curve of "Speed—R.P.M." should be established, besides that at highest speed. These additional points should be so spaced as to cover the curve from about half power of the engines to

full power. Approximately equal increments should be maintained in the power developed between the different series of runs. Each of these points on the curve should be established by at least three runs.

In conducting a trial of this character so as to get other information besides the contract speed, additional points should be determined on the curve, carrying it to as low a speed as possible; the initial friction of the engine should be determined; and, in general, all data bearing on the performance of the vessel and its machinery collected, so as to determine the various elements of efficiency throughout as wide a range of speeds and powers as possible. To a trial of such a character I do not wish to refer at the present time, as going beyond the subject immediately under discussion.

There is one point, however, upon which I would like to lay special emphasis, and that is the necessity of extreme care in the collection of data as to horsepower, if the result is to be worthy serious consideration. The use of a type of indicator with a large piston area, so as to minimize the relative importance of its internal friction; the use of external springs so that the spring may not be exposed either directly to the steam or to the radiation from the walls of the indicator body; these and similar precautions must all be taken if the observed results as to power are to be any real indication of the performance of the machinery. And although the amount of horsepower developed is not a contract requirement, yet the necessity of knowing that the conditions of performance of the machinery were satisfactory, implies the necessity of making the determination of the power developed with every reasonable precaution to insure accuracy.

From the foregoing it will be seen that only two important sources of probable error remain, after proper steps have been taken to obtain accurate data on the runs over the measured-mile course.

The first source of error is that due to variations in the personal errors of the observers who note the times on the mile. This can be partially eliminated by using the results

of the observations of three or more observers, and discarding all widely divergent records of observation before taking the average value of the remainder.

The second source of error is that due to irregular fluctuation in the strength of the current, and its effect may be reduced to a minimum by making repeated runs over the course. As an illustration, the speed of a vessel as determined by three runs over a mile course, required 125.6 R.P.M. to make the contract speed; a second series of three runs indicated 126 R. P.M. to be necessary. The average of the results of the two groups gave 125.8 R.P.M. as that necessary to maintain the contract speed. For this region of the mean speed curve, as established by the average of the two groups of runs, a given number of R.P.M. would indicate a speed that would differ by 0.035 of a knot from the speeds indicated by the two curves as established by the first or second groups of three runs taken singly.

With proper precautions for obtaining accuracy in the observations taken during a standardization trial, it is believed that the probable error in the speed so determined will not exceed .05 of a knot.

When the results of the standardization trial are applied to the observations made on a four-hour run at sea, an error is introduced into the speed so obtained, owing to the difference in the state of the wind and sea at the times the two trials are held. The resulting error is, however, as slight in amount as such a variation in the external conditions must necessarily cause with any method of trial.

It appears, therefore, that the "standardization of the screw" method of determining speed during contract trials is not only more equitable as concerns the interests of both the shipbuilder and the Government, but is also more accurate than any other method as yet proposed.

In conclusion, I wish to express my indebtedness and thanks to Professors C. H. Peabody and G. V. Wendell, of the Massachusetts Institute of Technology, for assistance in the preparation of this paper; and to Messrs. J. W. Clary and

C. A. Ford, of the Scientific Department of the Bureau of Construction and Repair, Navy Department, for their cordial co-operation in preparing the appended plates.

APPENDIX I.

Extract from Instructions Governing the Speed Trial of U. S. S. "Constellation" for Observing Vessels.

The run will begin at Station No. 1.

Tidal observations will be reduced to, and recorded in, hundredths of knots.

Time will be reduced to the 75th meridian.

Vessels will reach their stations (as designated upon chart) by 7 h. 30 m. o'clock A. M. of the day appointed for trial.

Vessels will take stations as follows:—

Station 1.—U. S. S. *Constitution.*

Station 2.—U. S. S. *Chesapeake.*

Etc.

All vessels will anchor so as to lie about 200 yards west of their respective stations, which are marked by buoys. They will carefully avoid fouling the buoy in any way, and will also avoid masking the buoy by tailing across the line.

Tidal observations will be begun at 8 h. 00 min. A. M. on the trial day, and will be made and recorded at fifteen-minute intervals (on even hours, half hours, and quarters of the 75th meridian time) until thirty minutes after the ship has passed the station on her return run; also at the approximate time of her passing station, allowing sufficient time either before or after exact passage to prevent log chip being affected by the wash.

Care will be taken in recording the tidal observations to enter as the direction of the current the point of the compass toward which the current sets.

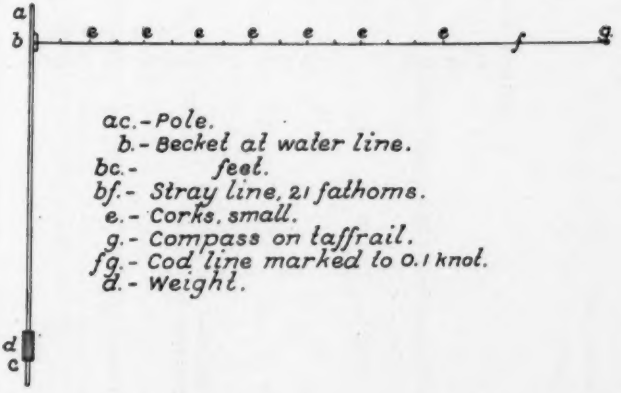
Taking the tidal observations is the most important duty of the station vessels, and can be done with accuracy only when they are motionless. Should, therefore, a vessel get into any

difficulties with her ground tackle and drag, or be obliged for any cause to employ her steam to move in the water, she will send a boat with tidal pole, compass, observer, and timepiece to moor to the buoy and take the observations there.

The following directions for taking current observations will be followed on the trial of the *Constellation*.

Directions for Current Observations.

(1) A pole 26 feet long, weighted to draw 24 feet. Diameter of pole, between 2½ and 3½ inches. A line of halliard stuff, about 15 fathoms long, buoyed by small corks about 1 fathom apart. To this line is hitched a good cod line. Leave about 6 fathoms of this cod line as additional stray line, making 21 fathoms stray line altogether. Mark the rest of the cod line to tenths of a knot; the length of a knot to be 47 feet 3 inches. The end of the halliard stuff is secured to the pole at water line.



(2) The cod line should be stretched and wetted before marking. The cod line should be measured and corrected before and after the trial, and correction noted. The drift of the line to be taken with a compass by observing the end of the pole, from the taffrail. Give magnetic direction, corrected for local deviation (if any) as per blanks.

(3) All current observations will be taken by a commissioned officer, who will sign the report certifying to the accuracy of the results given therein.

(4) The velocity of the current will be determined by measuring with a watch the number of seconds necessary for 94 feet 6 inches of the line to be paid out, or if this time is in excess of five minutes, the length of line paid out in five minutes will be noted. From these measurements the velocity of the current in knots per hour will be calculated and entered on the blank furnished for that purpose.

(5) A full copy will, however, also be appended, giving the exact record of observations as taken for both time and length of the line paid out, from which the velocity of the current was obtained.

* * * * * * * * * *

(6) Timepieces will be compared with a reliable chronometer, before and after taking observations.

* * * * * * * * * *

PAUL JONES,
Captain, U. S. Navy, President of Board.

Form in which Observation Vessels make their Reports.

U. S. S. *CONSTITUTION*, STATION 1. November 2, 1904.

Current and other Observations during the Speed Trial of the Constellation.

75th meridian time.	Current (magnetic).		Wind (magnetic).		Sea.	Remarks.
	Direction in degrees.	Force. Knots.	Direction.	Force.		
8 h. 00 m. A. M.	S. 85° E.	.046	N. 67° E.	2	Smooth	*Constellation* passed buoy at 8 h. 13 m. 30 s. A. M.
8 h. 15 m. A. M.	N. 43° E.	.018	N. 67° E.	2	Smooth	
Etc.	Etc.	Etc.	Etc.	Etc.	Etc.	

I certify to the correctness of the above observations.

JOHN JONES,
Lieutenant, U. S. Navy.

Very respectfully,

HENRY SMITH,
Captain, U. S. N.,
Commanding.

To Captain PAUL JONES, U. S. N.,
Senior Member Constellation Trial Board.

Table I.

Comparison of Performances on Official Speed Trials U. S. Monitors Arkansas, Nevada, Florida.

Name of Ship.		*Arkansas.**	*Nevada.†*	*Florida.†*
Speed of ship, knots per hour (during two-hour trial)		12.03	13.039	12.4
Propellers, pitch		7′ 3″	7′ 9.55″	8′ $3\frac{13}{16}$″
Propellers, diameter		9′	9′ 2.7″	8′ 6″
Revolutions, main engines	Starboard	197.9	186.46	203.1
	Port	196.3	188.19	198.4
	Mean	197.1	187.32½	200.75
Slip of propellers, per cent. (mean)		9.44	9.58	24.29
Steam pressure, at boilers (gauge)		204 lbs.	238.8 lbs.	203.5 lbs.
Steam pressure, at engines (gauge)		199.	188.7	191.35
Vacuum		25.8″	27.8″	26.2″
Collective I.H.P. main engines, air, circulating and feed pumps		1,738.78	1,969.77	2,335.68

* Standardization trial.

† Double run over 12-knot course.

† Double run over 12-knot course.

Table II.

Trial of the U. S. Battleship Missouri. Current Correction Sheet.

	Squares.	*T.* Knots or time in squares.	Strength of current, knots.	Direction of current.	Course, magnetic.	Keel angle.	Current in time *T.*	Distance in squares.	Distance corrected.	Sum of corrected distances.	Sum of elapsed times.	Speed.	Total of all elapsed times.	Total of all corrected distances.	Final corrected speed.
		m. s.		*degs.*	*degs.*	*aegs.*					*h. m. s.*		*h. m. s.*	*miles.*	*knots.*
Run North	1	11 9.5	.422	S. 16 E.	N. 22 E.	142	.0785	3.3	3.3622						
	2	22 19.25	.614	S. 17 W.	"	175	.2284	6.6	6.8275						
	3	22 36.25	.549	South.	"	158	.2068	6.6	6.7922						
	4	22 36.75	.868	S. 45 W.	"	157	.3271	6.6	6.9024						
	5	22 22.	.172	S. 12 E.	"	146	.0641	6 6	6.6533						
	6	11 11.75	.099	S. 21 E.	"	137	.0185	3 3	3.3135	33.8511	1 52 15.5	18.0927			
Run South	7	10 57.25	.055	S. 29 E.	S. 22 W.	51	.0100	3.3	3.2937						
	8	21 31.5	.189	S. 8 W.	"	14	.0678	6.6	6.5342						
	9	21 08.	.470	S. 40 W.	"	18	.1655	6.6	6.4427						
	10	21 09.75	.307	S. 20 W.	"	2	.1083	6.6	6.4918						
	11	21 29.25	.400	S. 12 W.	"	10	.1433	6.6	6.4589						
	12	10 53.25	.227	S. 35 E.	"	57	.0412	3.3	3.2777	32.4990	1 47 0.9	18.1982			
													3 39 24.5	66.3501	18.1443

Table III.

Summary of Certain Speed Trials.

Name of ship.	*Missouri.*	*Denver.*	*Colorado.*	*West Virginia.*	*Pennsylvania.*
Run North.					
Passed buoy south end course	h. m. s. 11 13 12.5	h. m. s. 10 46 47.5	h. m. s. 8 45 6.0	h. m. s. 9 56 40.5	h. m. s. 8 52 11.0
Passed buoy north end course	1 5 28.0	12 46 2.0	10 45 21.0	11 55 39.5	10 51 34.0
Elapsed time	1 52 15.5	1 59 14.5	2 0 15.0	1 58 59.0	1 59 23.0
Distance uncorrected (knots)	33.0	33.0	44.019	44.019	44.019
Speed uncorrected (knots per hour)	18.1721	16.6046	21.9634	22.1971	22.1234
Distance corrected for current (knots)	33.8511	32.9486	44.1995	43.7336	44.3701
Speed corrected (knots per hour)	18.0927	16.5791	22.054	22.054	22.300
Difference between corrected and uncorrected speeds (knots per hour)	(−) 0.0794	(−) 0.0255	(+) 0.0906	(−) 0.1431	(+) 0.1766
Revolutions per minute, starboard engine	117.23	200.7687	124.8297	128.5	125.9
Revolutions per minute, port engine	117.62	202.5379	126.7205	124.2	126.9
Revolutions per minute, mean	117.425	201.6533	125.7751	126.35	126.4
Run South.					
Passed buoy north end course	h. m. s. 1 23 14.5	h. m. s. 2 6 23.5	h. m. s. 10 58 30.0	h. m. s. 12 13 18.0	h. m. s. 11 3 31.5
Passed buoy south end course	3 10 23.5	4 10 33.0	12 55 22.0	2 12 52.0	12 59 37.5
Elapsed time	1 47 9.0	2 4 9.5	1 56 52.0	1 59 34.0	1 56 6.0
Distance uncorrected (knots)	33.0	33.0	44.019	44.019	44.019
Speed uncorrected (knots per hour)	18.4791	15.9474	22.5993	22.0890	22.7488
Distance corrected for current (knots)	32.499	33.7465	43.7075	44.3134	43.6835
Speed corrected (knots per hour)	18.1982	16.3081	22.44	22.237	22.575
Difference between corrected and uncorrected speeds (knots per hour)	(−) 0.2809	(+) 0.3607	(−) 0.1593	(+) 0.1480	(−) 0.1738
Revolutions per minute, starboard engine	116.00	195.4763	127.7957	130.9	129.9
Revolutions per minute, port engine	117.02	197.9570	128.1408	126.3	128.9
Revolutions per minute, mean	116.51	196.7167	127.9683	128.6	129.4

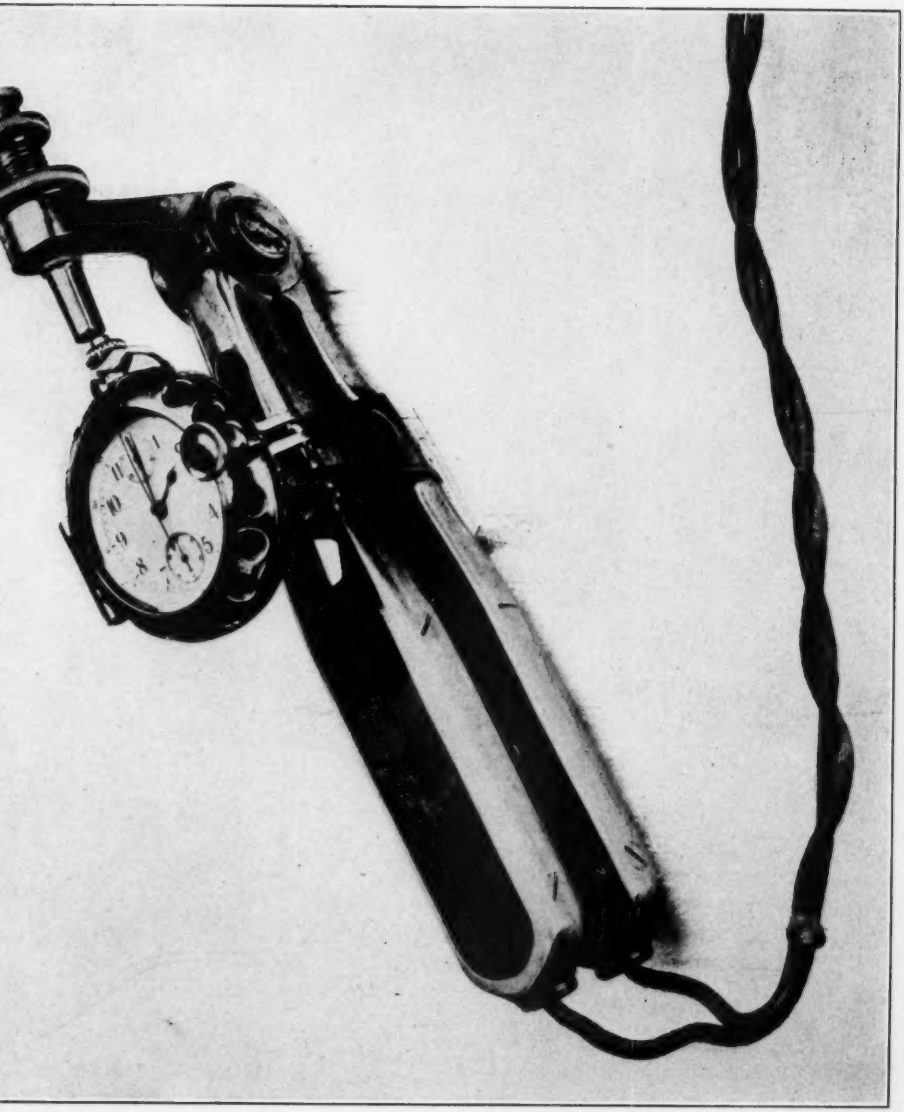
Average of Runs.					
Corrected distance for entire run	66.3501	66.6951	87.9070	88.047	88.0536
Sum of elapsed times	h. m. s. 3 39 24.5	h. m. s. 4 3 24.0	h. m. s. 3 57 7.0	h. m. s. 3 58 33.0	h. m. s. 3 55 29.0
Corrected speed (knots per hour)	18.1443	16.4409	22.244	22.146	22.436
Uncorrected speed for entire run (knots per hour)	18.3256	16.2760	22.2814	22.1431	22.4361
Difference between corrected and uncorrected speeds (knots per hour)	(—) 0.1813	(+) 0.1649	(—) 0.0374	(+) 0.0029	(—) 0.0001
Revolutions per minute entire run, starboard engine	116.615	198.1225	126.3127	129.7	127.9
Revolutions per minute entire run, port engine	117.32	200.2475	127.4307	125.25	127.9
Revolutions per minute entire run, mean	116.9675	199.1850	126.8717	127.475	127.9
I.H.P. main engines	15,642.1	5,558.66	26,153.95	25,726.1	28,006.1
I.H.P. main engines, air, circulating, hot-well, and feed pumps	15,860.78	5,633.16	26,845.68	26,134.83	28,618.41
I.H.P. total all machinery	16,276.67	5,686.90	27,373.83	26,465.57	29,070.95

Table IIIa.

Summary of Speed Trials—Time of Passing Station Buoys.

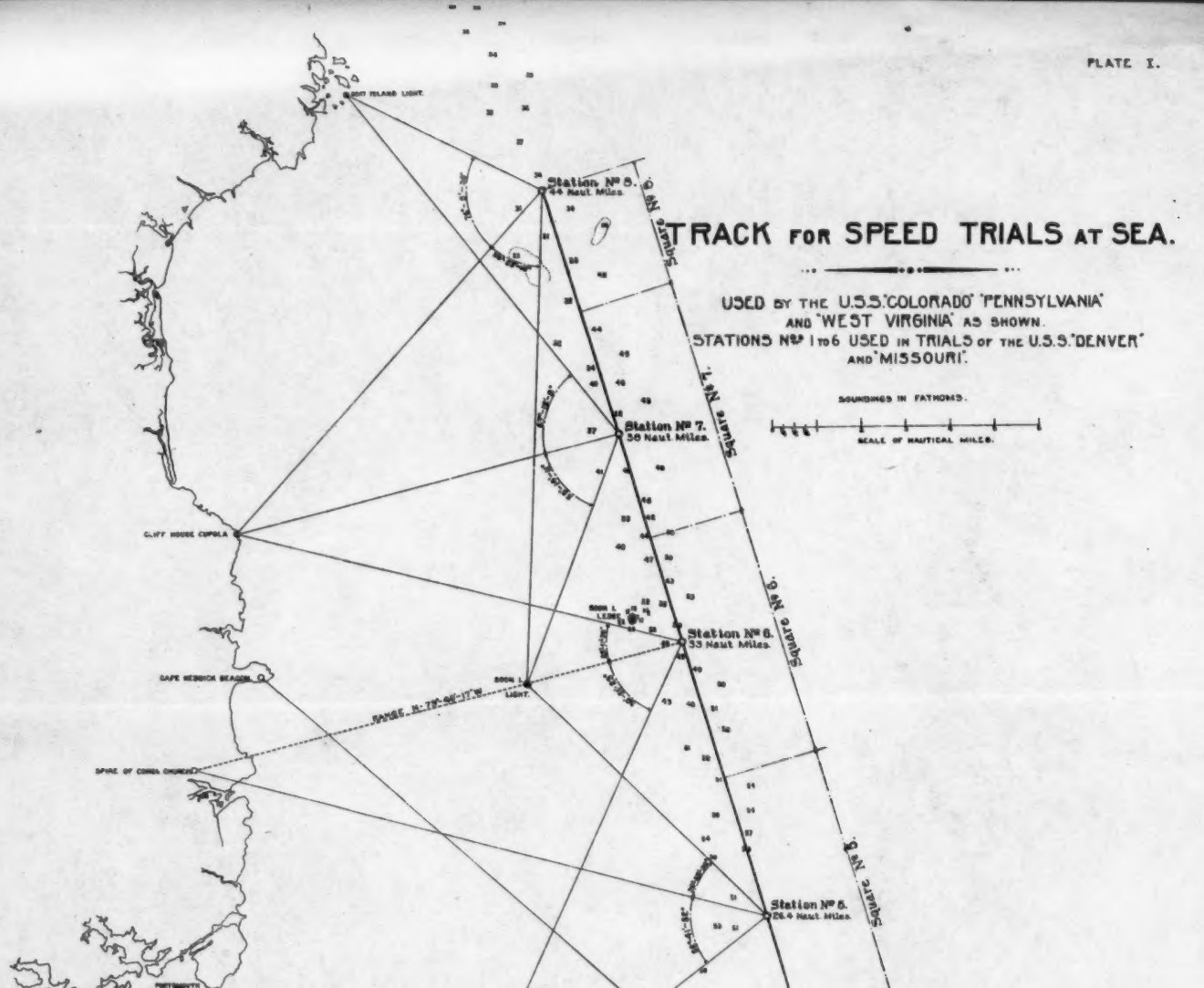
Name of ship.	*Missouri.*		*Denver.*		*Colorado.*		*West Virginia.*		*Pennsylvania.*	
Date of trial.	October 21, 1903.		October 22, 1903.		October 24, 1904.		November 2, 1904.		November 23, 1904.	
	Time passing buoy.	Time in "squares."	Time passing buoy.	Time in "squares."	Time passing buoy.	Time in "squares."	Time passing buoy.	Time in "squares."	Time passing buoy.	Time in "squares."
Run North.	*h. m. s.*	*m. s.*	*h. m. s.*	*m. s.*	*h. m. s.*	*m. s*	*h. m. s.*	*m. s*	*h. m. s.*	*m. s.*
Station No. 1...	11 13 12.5	11 9.5	10 46 47.5	11 51.75	8 45 6.	9 1.75	9 56 40.5	8 55.	8 52 11.	9 4.25
" 2...	11 35 31.625	22 19.25	11 10 32.	23 45.5	9 3 12.25	18 9.	10 14 32.625	17 54.25	9 10 20.375	18 10.25
" 3...	11 57 59.375	22 36.25	11 34 29.25	24 9.	9 21 24.625	18 15.75	10 32 22.875	17 46.25	9 28 22.6875	26 59.5
" 4...	12 20 35.875	22 36.75	11 58 35.625	24 3.75	9 39 36.25	18 7.5	10 50 4.875	17 37.75	9 46 25.	.....
" 5...	12 43 5.250	22 22.	12 22 25.75	23 36.5	9 57 40.625	18 1.25	11 7 47.625	17 47.75	10 3 41.	26 42.25
" 6...	1 5 28.	11 11.75	12 46 2.	11 48.	10 14 34.	15 45.5	11 24 34.	15 45.	10 20 57.	15 39.5
" 7...	.....	.....	.....	.....	10 29 51.125	14 48.75	11 39 56.75	15 00.5	10 35 8.25	14 43.
" 8...	.....	.....	.....	.....	10 45 21.	8 5.5	11 55 39.5	8 12.5	10 51 34.	8 4.25
Run South.										
Station No. 8...	.....	.....	.....	.....	10 58 30.	8 4.	12 13 18.	8 1.5	11 3 31.5	7 58.25
" 7...	.....	.....	.....	.....	11 13 56.625	14 45.25	12 28 42.625	14 46.25	11 18 48.	14 36 5
" 6...	1 23 14.5	10 57.25	2 6 23.5	12 40.	11 29 4.5	15 30.5	12 43 58.625	15 45.75	11 33 49.	15 25.5
" 5...	1 44 57.5	21 31.5	2 31 26.125	24 45.25	11 45 33.375	17 27.25	1 0 48.375	17 53.75	11 50 41.	26 1.25
" 4...	2 6 7.25	21 8.	2 55 58.125	24 18.75	12 2 51.125	17 8.25	1 18 38.375	17 46.25	12 7 33.	.....
" 3...	2 27 26.125	21 9.75	3 20 23.375	24 31.75	12 20 7.5	17 24.5	1 36 32.875	18 2.75	12 24 54.6875	26 0.25
" 2...	2 48 45.625	21 29.25	3 45 12.25	25 6.	12 37 41.375	17 43.25	1 54 41.	18 13.5	12 42 16.375	17 26.25
" 1...	3 10 23.5	10 53.25	4 10 33.	12 47.75	12 55 22.	8 49.	2 12 52.	9 4.25	12 59 37.5	8 38.

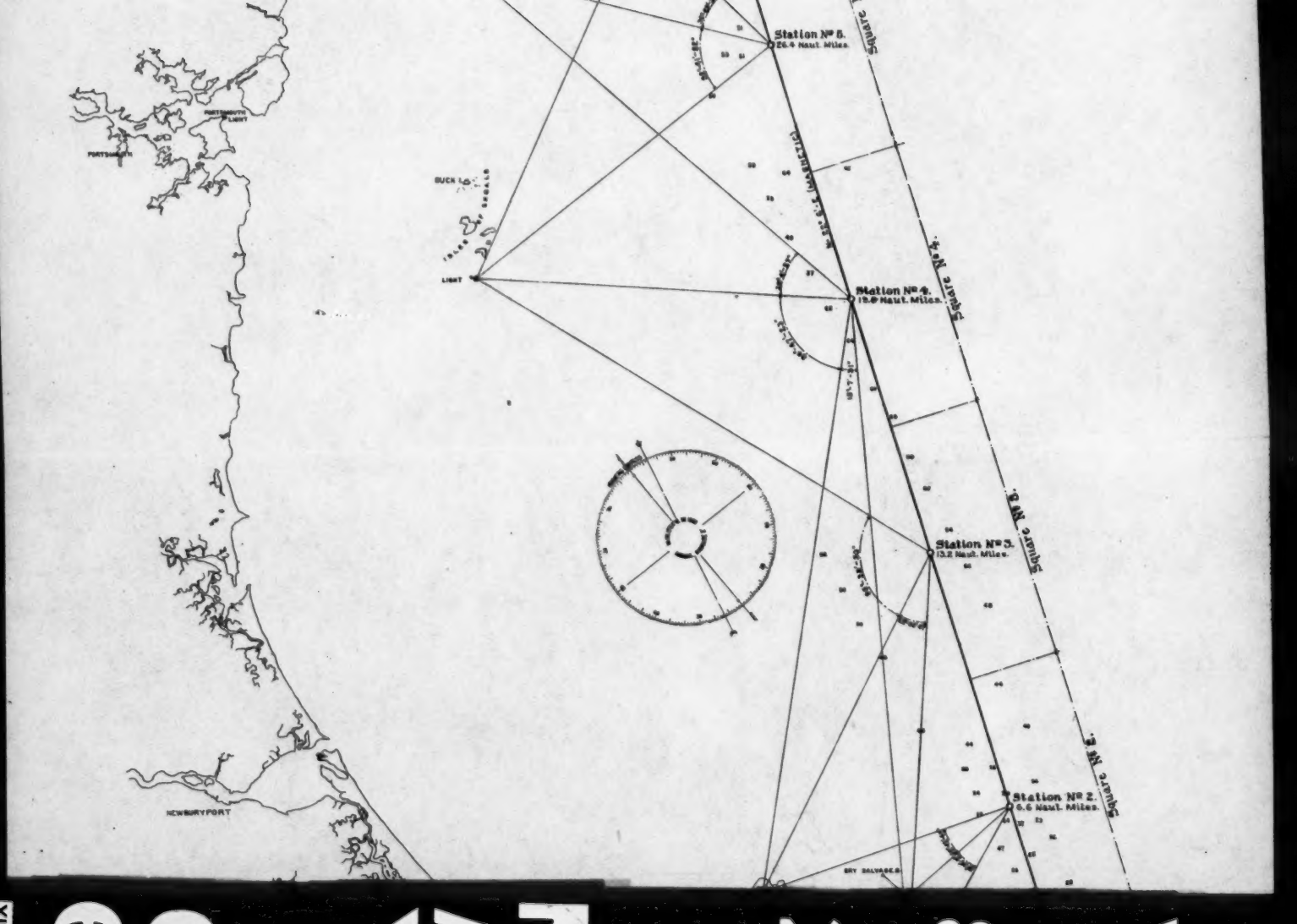
	2...	2	48	45.625	21	29.25	3	45	12.25	25	0.	12	37	41.375	17	43.25	1	54	41.	18	13.5	12	42	16.375	17	28.25
"	1...	3	10	23.5	10	53.25	4	10	33.	12	47.75	12	55	22.	8	49.	2	12	52.	9	4.25	12	59	37.5	8	38.

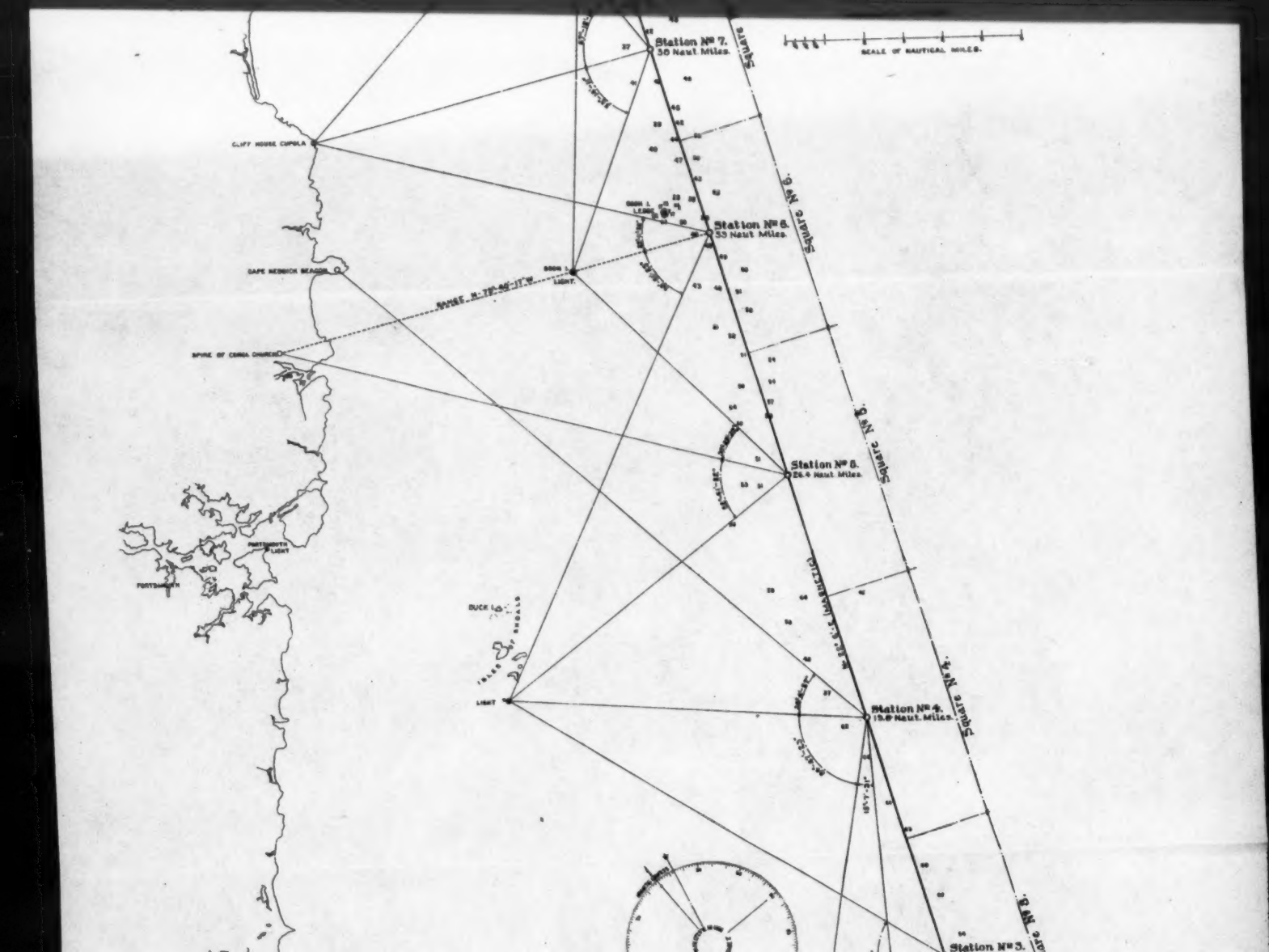


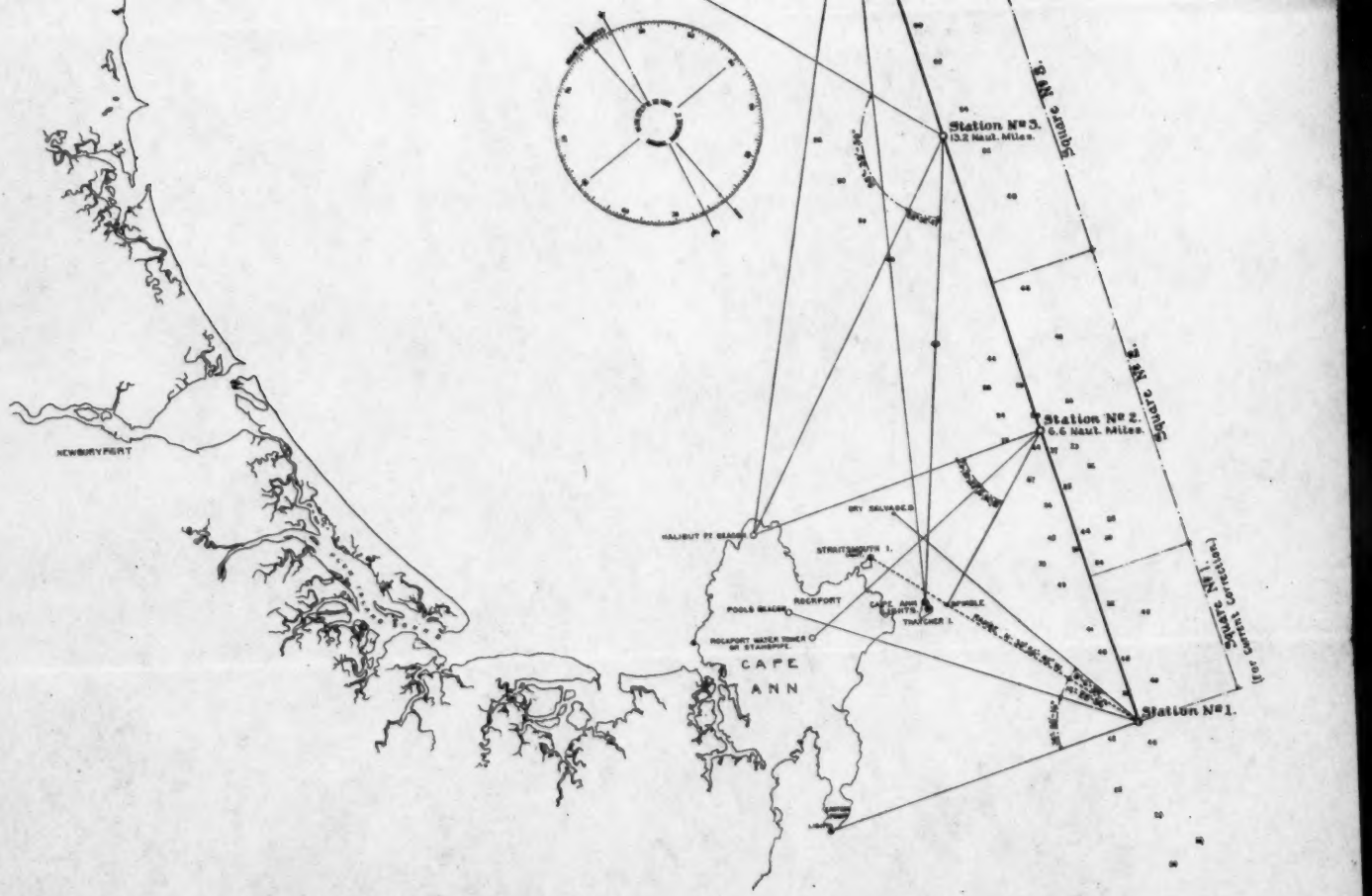
STOP WATCH CONNECTION FOR MARKING TIME.

PLATE I.

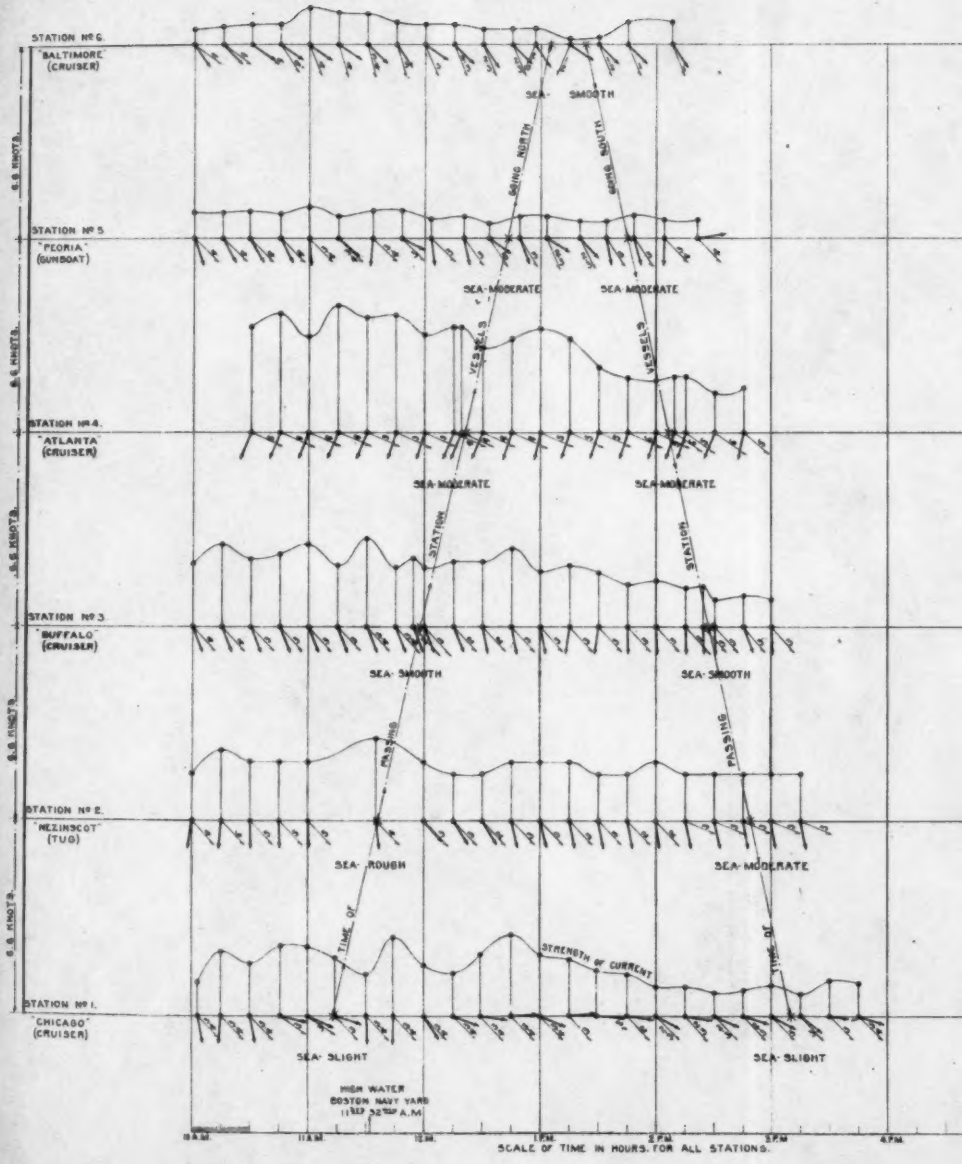

TRACK FOR SPEED TRIALS AT SEA.
USED BY THE U.S.S. "COLORADO" "PENNSYLVANIA" AND "WEST VIRGINIA" AS SHOWN.
STATIONS Nos 1 TO 6 USED IN TRIALS OF THE U.S.S. "DENVER" AND "MISSOURI".
SOUNDINGS IN FATHOMS.
SCALE OF NAUTICAL MILES.
Station No 5. 44 Naut Miles.
Station No 7. 38 Naut Miles.
Station No 6. 33 Naut Miles.
Station No 8. 28.4 Naut Miles.
Square No 5.
Square No 7.
Square No 6.
Square No 8.
RANGE N. 73° 06' 17" W.
BOON I. LIGHT.
CLIFF HOUSE CUPOLA
CAPE HENRICK BEACON
SPIRE OF CHINA CHURCH


Station Nº 5.
26.4 Naut. Miles.
Station Nº 4.
19.8 Naut. Miles.
Station Nº 3.
13.2 Naut. Miles.
Station Nº 2.
6.6 Naut. Miles.
Square Nº 4.
Square Nº 3.
Square Nº 2.
PORTSMOUTH
PORTSMOUTH LIGHT
NEWBURYPORT
BUCK I.
LIGHT
DRY SALVAGE


Station No 7.
36 Naut. Miles.
Station No 6.
33 Naut Miles.
Station No 5.
26.4 Naut Miles.
Station No 4.
19.8 Naut. Miles.
Square No 6.
Square No 5.
Square No 4.
SCALE OF NAUTICAL MILES.
CLIFF HOUSE CUPOLA
CAPE NEDDICK BEACON
SPIRE OF CONGA CHURCH
PORTSMOUTH
PORTSMOUTH LIGHT
BUCK I.
LIGHT


Station No 3.
13.2 Naut. Miles.
Square NE.
Station No 2.
6.6 Naut. Miles.
Square NE.
Square 2 NE. (Intermediate Point)
Station No 1.
NEWBURYPORT
HALIBUT PT. BEACON
DRY SALVAGE I.
STRAITSMOUTH I.
ROCKPORT
POOLS BEACH
ROCKPORT WATER TOWER OR STANDPIPE
CAPE ANN LIGHTS
THACHER I.
CAPE ANN

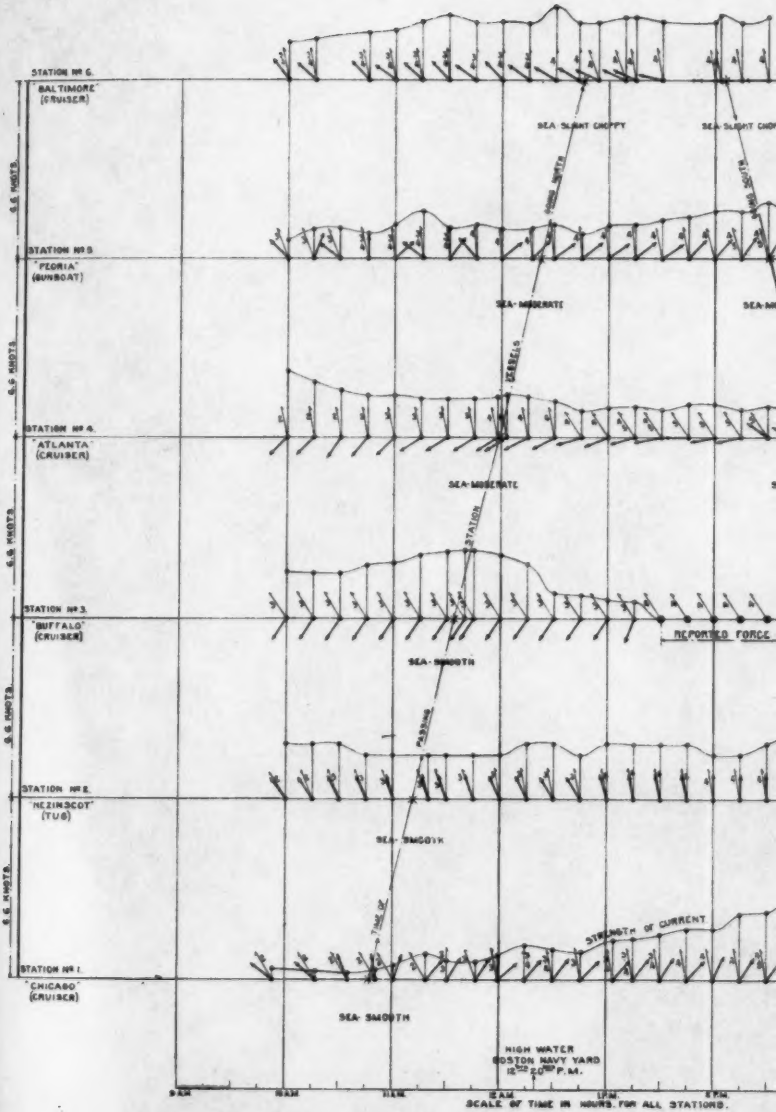
CURRENT AND OTHER OBSERVATIONS
TAKEN
DURING SPEED TRIAL
OF THE
U.S.S. MISSOURI.
OCT. 21st 1903.



NOTE.-
——→ DIRECTION OF CURRENT RELATIVE TO COURSE.
- - -→ DIRECTION OF WIND RELATIVE TO COURSE AND FORCE BY BEAUFORT SCALE.

PLATE II.

CURRENT AND OTHER OBSERVATIONS
TAKEN
DURING SPEED TRIAL
OF THE
U.S.S. DENVER.
OCT. 22ND 1903.



NOTE:-
——→ DIRECTION OF CURRENT RELATIVE TO COURSE.
—N→ DIRECTION OF WIND RELATIVE TO COURSE AND FORCE BY BEAUFORT SCALE.

ATIONS

PLATE III.

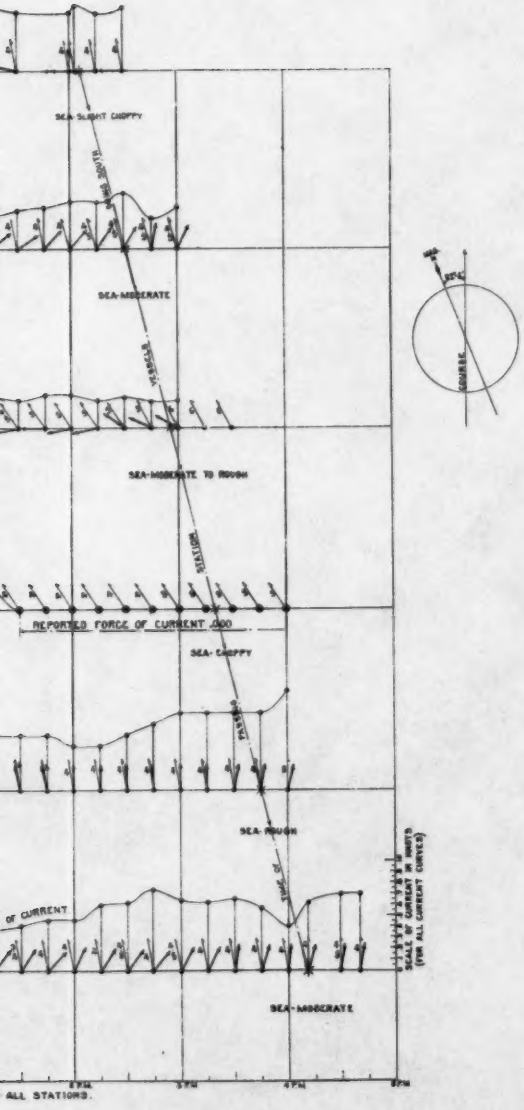
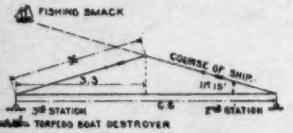


FIG. 1.



$$x = \frac{3.3}{\cos 11°15'} = 3.3646$$

ADDITIONAL DISTANCE RUN = 2(3.3646 − 3.3) = .1292

FIG. 2.

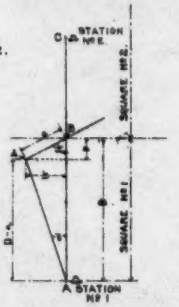
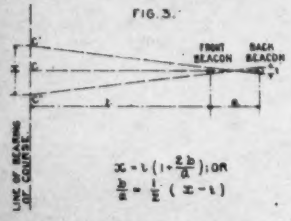


FIG. 3.



$$x = l\left(1 + \frac{2b}{a}\right) : \text{OR}$$

$$\frac{b}{a} = \frac{1}{2}(x - 1)$$

FIG. 7.

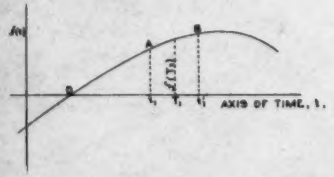
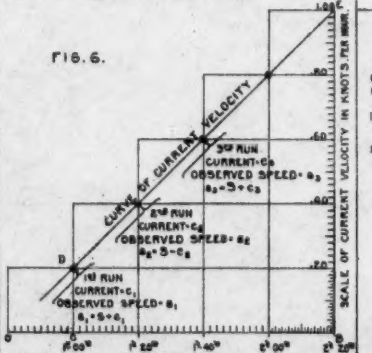


FIG. 6.



TRUE SPEED OF VESSEL = $S = \frac{1}{2}(s_1 + 2s_2 + s_3)$

1st METHOD

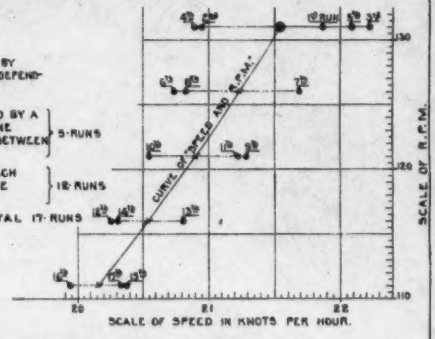
CURVE OF SPEED AND R.P.M. ESTABLISHED BY FIVE POINTS, EACH DETERMINED BY AN INDEPENDENT SERIES OF RUNS.

1° POINT AT HIGHEST SPEED ESTABLISHED BY A SERIES OF FIVE RUNS AT SAME ENGINE SPEED, WITH EQUAL TIME INTERVALS BETWEEN THE RUNS. } 5 RUNS

2° FOUR POINTS AT LOWER SPEEDS, EACH ESTABLISHED BY A SERIES OF THREE RUNS, MADE AS IN 1° } 12 RUNS

TOTAL 17 RUNS

FIG. 5



2nd METHOD

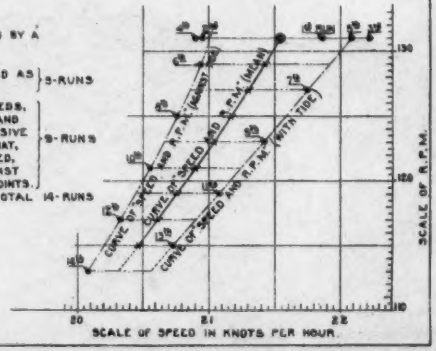
CURVE OF SPEED AND R.P.M. ESTABLISHED BY A CONTINUOUS SERIES OF RUNS.

1° POINT AT HIGHEST SPEED ESTABLISHED AS IN 1st METHOD. } 5 RUNS

2° ADDITIONAL RUNS MADE AT LOWER SPEEDS, WITH EQUAL INTERVALS OF BOTH TIME AND ENGINE REVOLUTIONS BETWEEN SUCCESSIVE RUNS. NUMBER OF RUNS TO BE SUCH THAT, INCLUDING THE FIFTH RUN AT TOP SPEED, EACH CURVE OF RUNS WITH AND AGAINST TIDE WILL BE ESTABLISHED BY FIVE POINTS. } 9 RUNS

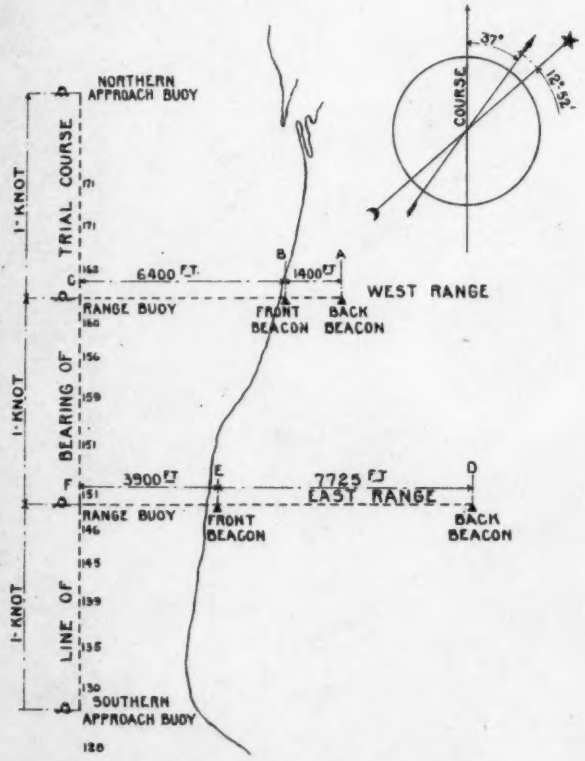
TOTAL 14 RUNS

FIG. 9.


CURVE OF SPEED AND R.P.M. (MEAN)
CURVE OF SPEED AND R.P.M. (AGAINST TIDE)
CURVE OF SPEED AND R.P.M. (WITH TIDE)
SCALE OF R.P.M.
SCALE OF SPEED IN KNOTS PER HOUR.

TRIAL COURSE AT PROVINCETOWN MASS.

FIG. 4.

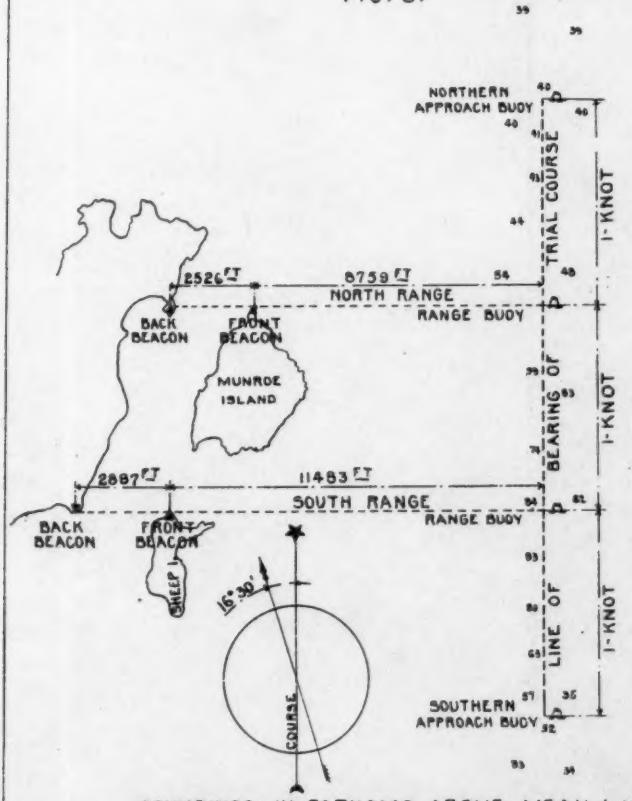


SOUNDINGS IN FEET ABOVE MEAN L.W.

MASS.

TRIAL COURSE AT ROCKLAND ME.

FIG. 5.



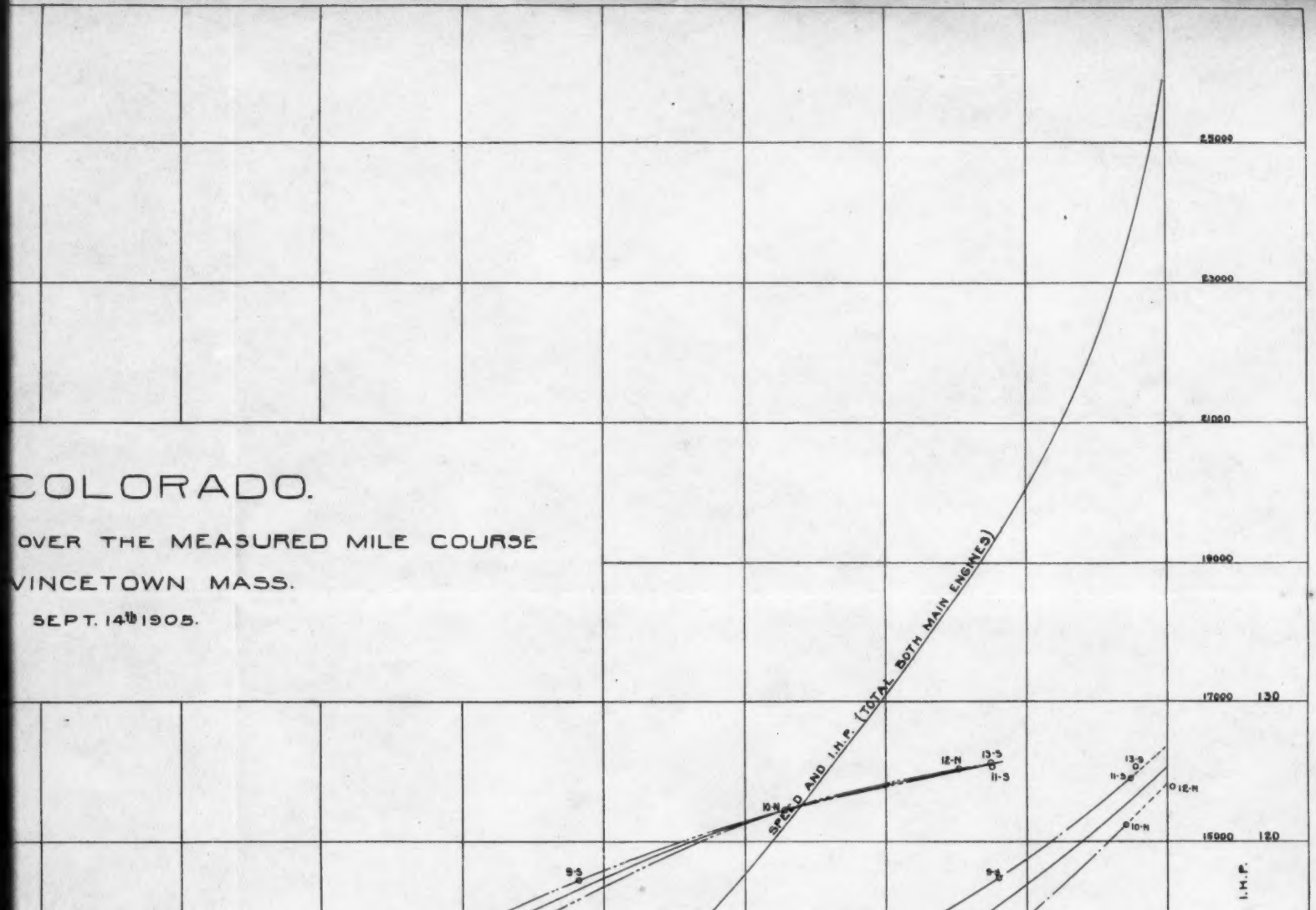
SOUNDINGS IN FATHOMS ABOVE MEAN L.W.

COLORADO.

OVER THE MEASURED MILE COURSE

VINCETOWN MASS.

SEPT. 14th 1905.



SCALE OF I.H.P. FOR CURVE OF SPEED AND I.H.P.

SCALE OF REV. PER MIN.

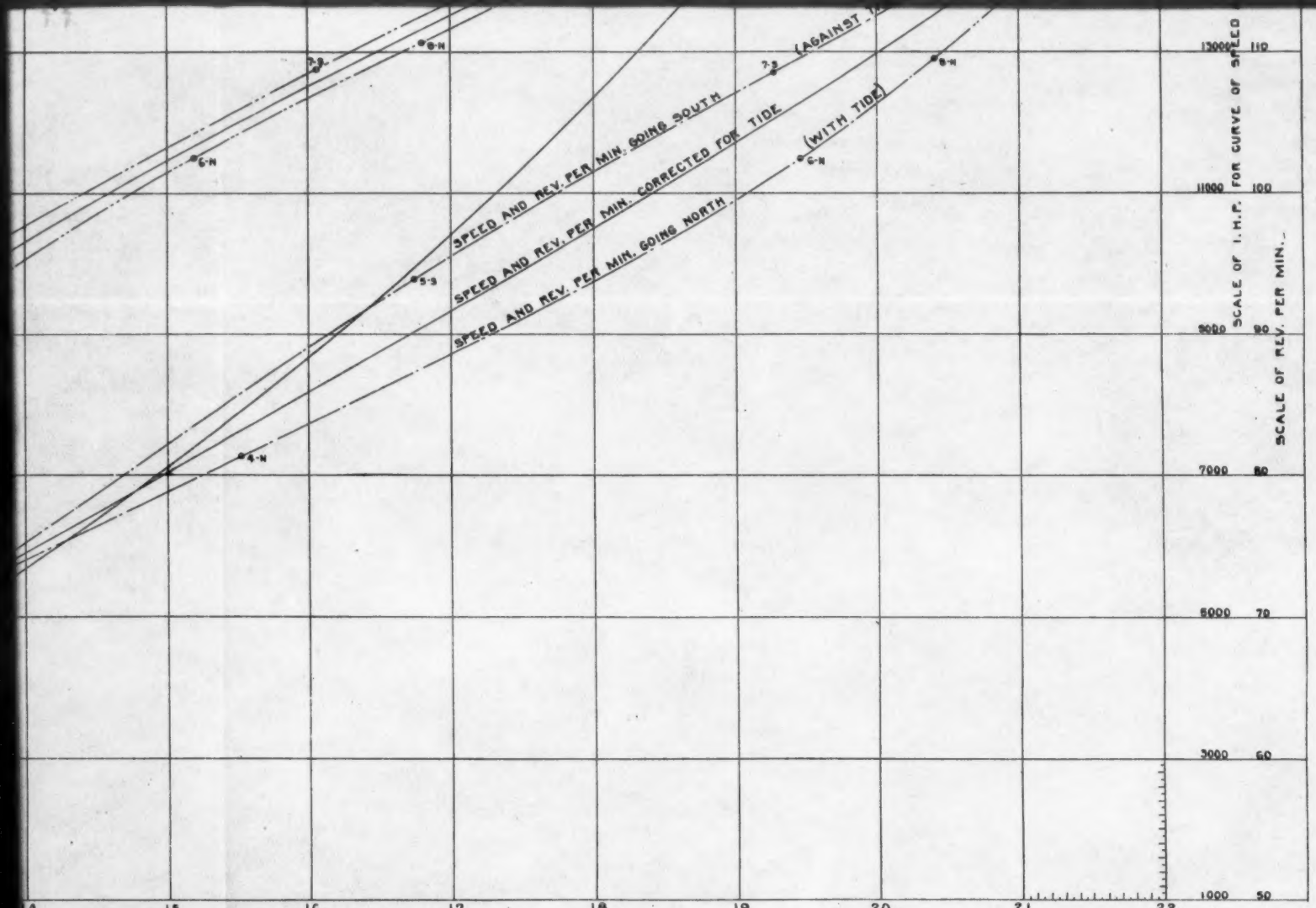
SPEED AND REV. PER MIN. GOING NORTH

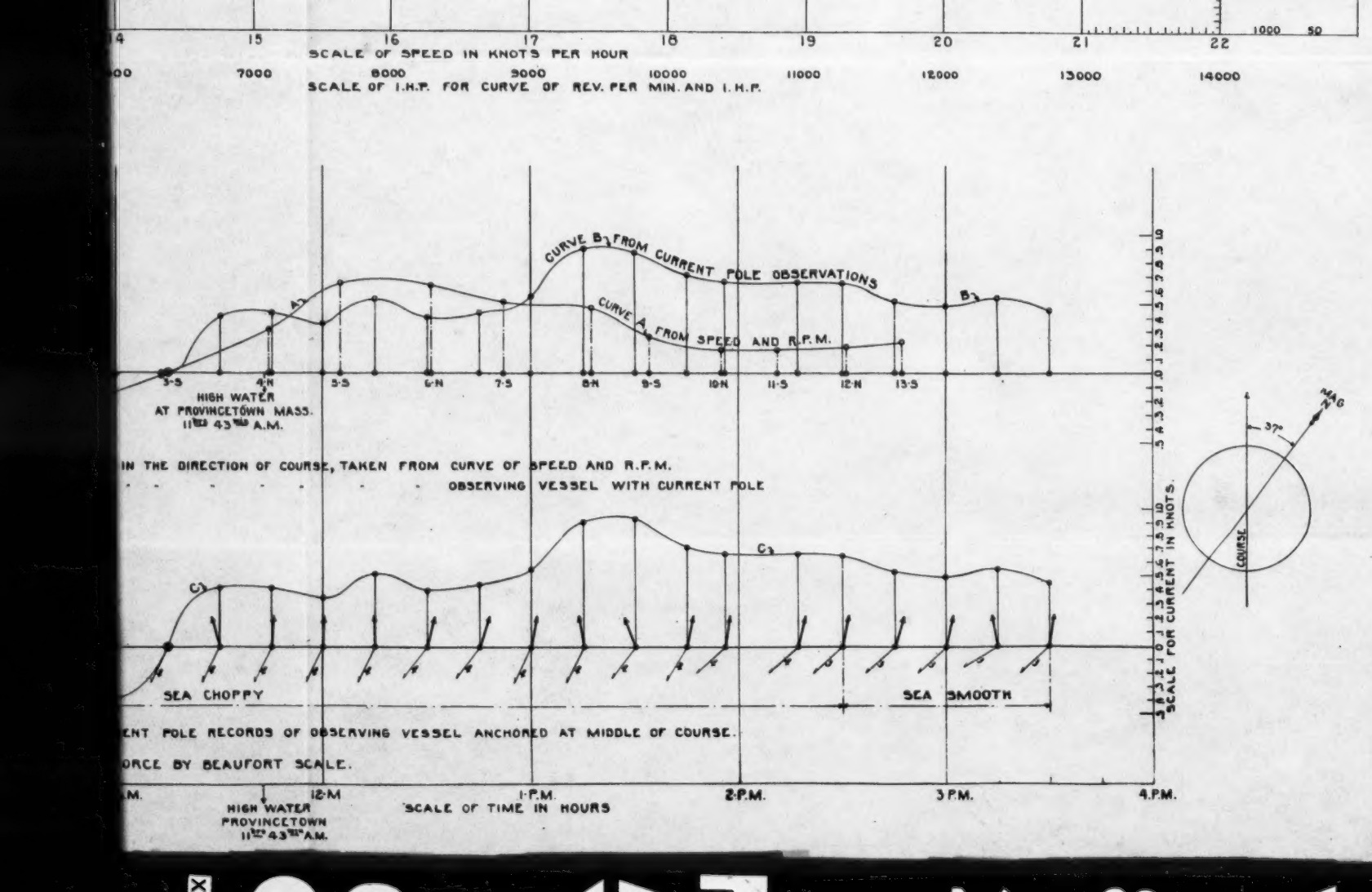
SPEED AND REV. PER MIN. CORRECTED FOR TIDE

SPEED AND REV. PER MIN. GOING SOUTH

(WITH TIDE)

(AGAINST TIDE)


SPEED AND REV. PER MIN. GOING SOUTH
SPEED AND REV. PER MIN. CORRECTED FOR TIDE
SPEED AND REV. PER MIN. GOING NORTH
(AGAINST TIDE)
(WITH TIDE)
7-S
6-N
5-S
4-N
SCALE OF I.H.P. FOR CURVE OF SPEED
SCALE OF REV. PER MIN.
13000 110
11000 100
9000 90
7000 80
5000 70
3000 60
1000 50
14 15 16 17 18 19 20 21 22
SCALE OF SPEED IN KNOTS PER HOUR
00 7000 8000 9000 10000 11000 12000 13000 14000
SCALE OF I.H.P. FOR CURVE OF REV. PER MIN. AND I.H.P.

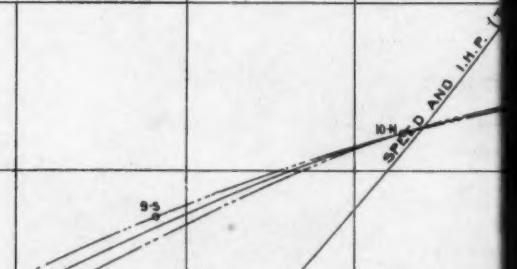

SCALE OF SPEED IN KNOTS PER HOUR
SCALE OF I.H.P. FOR CURVE OF REV. PER MIN. AND I.H.P.
CURVE B FROM CURRENT POLE OBSERVATIONS
CURVE A FROM SPEED AND R.P.M.
HIGH WATER AT PROVINCETOWN MASS. 11H 43M A.M.
IN THE DIRECTION OF COURSE, TAKEN FROM CURVE OF SPEED AND R.P.M.
OBSERVING VESSEL WITH CURRENT POLE
SEA CHOPPY
SEA SMOOTH
ENT POLE RECORDS OF OBSERVING VESSEL ANCHORED AT MIDDLE OF COURSE.
ORCE BY BEAUFORT SCALE.
SCALE FOR CURRENT IN KNOTS.
MAG
37°
COURSE
HIGH WATER PROVINCETOWN 11H 43M A.M.
SCALE OF TIME IN HOURS
12 M
1 P.M.
2 P.M.
3 P.M.
4 P.M.

U.S.S. COLORADO.

STANDARDIZATION TRIAL OVER THE MEASURED MILE COURSE

PROVINCETOWN MASS.

SEPT. 14th 1905.



NOTE CURVES OF REV. PER MIN. AND I.H.P. FOR STARBOARD ENGINE ONLY.

REV. PER MIN. AND I.H.P. (GOING SOUTH.)

REV. PER MIN. AND I.H.P. MEAN.

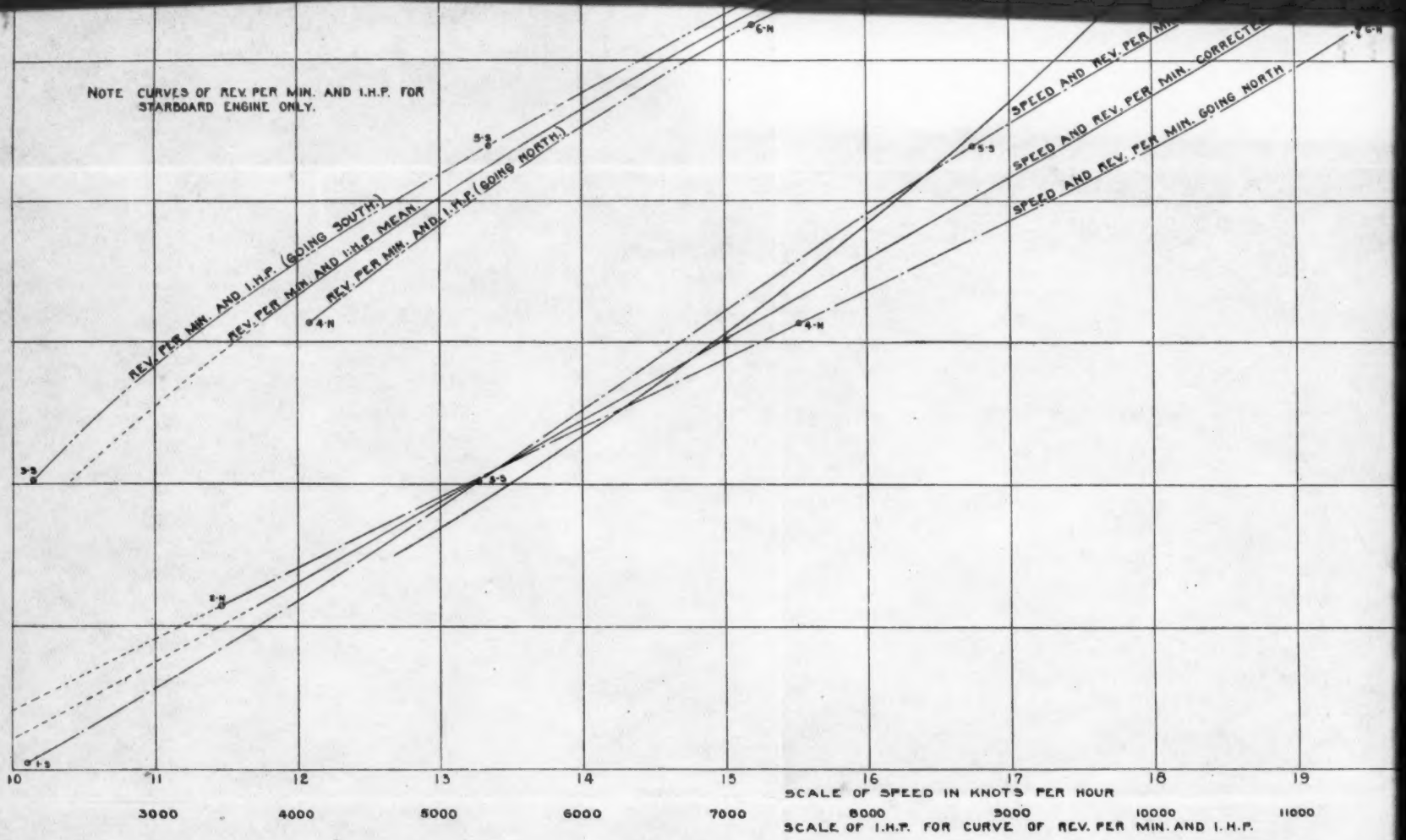
REV. PER MIN. AND I.H.P. (GOING NORTH.)

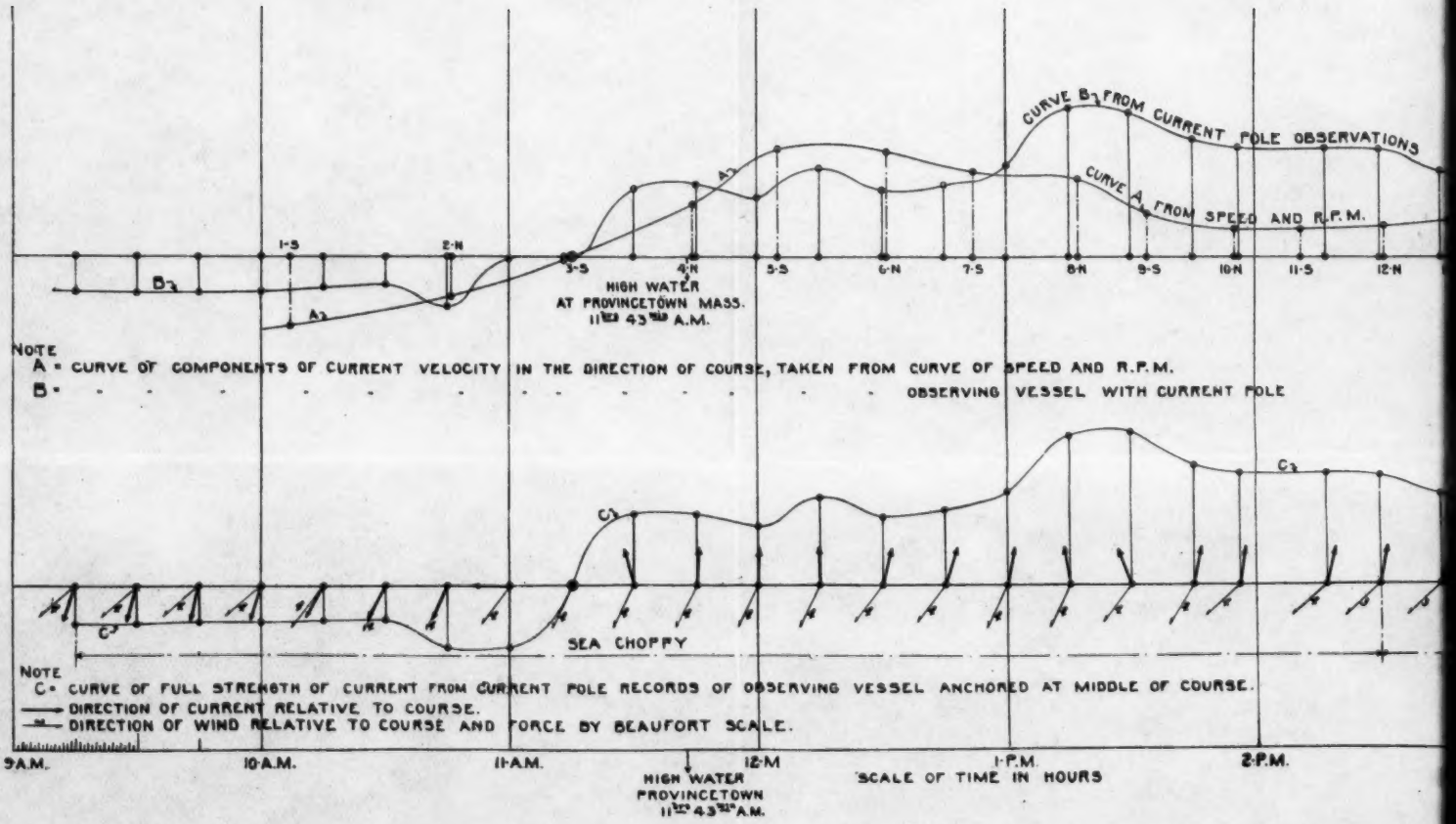
SPEED AND REV. PER MIN. CORRECTED FOR TIDE (WITH T

SPEED AND REV. PER MIN. GOING SOUTH

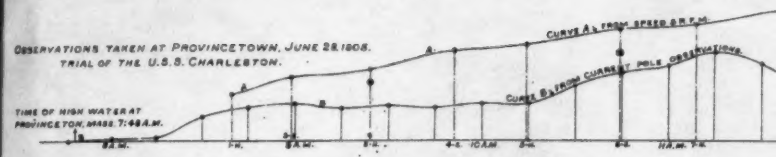
SPEED AND REV. PER MIN. GOING NORTH

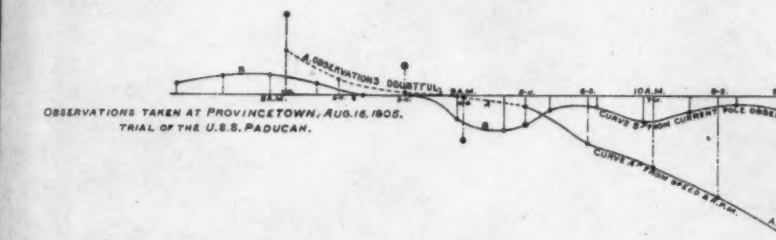
(AGAINST

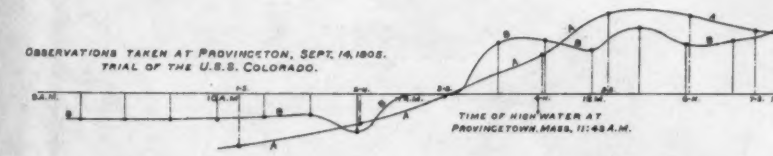

NOTE CURVES OF REV. PER MIN. AND I.H.P. FOR STARBOARD ENGINE ONLY.
REV. PER MIN. AND I.H.P. (GOING SOUTH.)
REV. PER MIN. AND I.H.P. MEAN.
REV. PER MIN. AND I.H.P. (GOING NORTH.)
SPEED AND REV. PER MIN.
SPEED AND REV. PER MIN. CORRECTED
SPEED AND REV. PER MIN. GOING NORTH
1-S
2-N
3-S
4-N
5-S
6-N
10
11
12
13
14
15
16
17
18
19
SCALE OF SPEED IN KNOTS PER HOUR
3000
4000
5000
6000
7000
8000
9000
10000
11000
SCALE OF I.H.P. FOR CURVE OF REV. PER MIN. AND I.H.P.

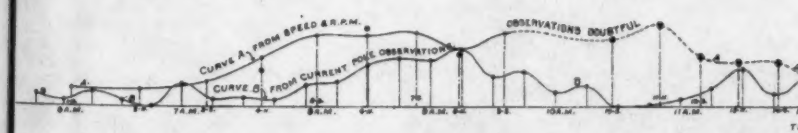

CURVE B FROM CURRENT POLE OBSERVATIONS
CURVE A FROM SPEED AND R.P.M.
B
A
1-S
2-N
3-S
4-N
5-S
6-N
7-S
8-N
9-S
10-N
11-S
12-N
HIGH WATER
AT PROVINCETOWN MASS.
11h 43m A.M.
NOTE
A = CURVE OF COMPONENTS OF CURRENT VELOCITY IN THE DIRECTION OF COURSE, TAKEN FROM CURVE OF SPEED AND R.P.M.
B = - - - - - - - - - - - - - - - - - - OBSERVING VESSEL WITH CURRENT POLE
C
SEA CHOPPY
NOTE
C = CURVE OF FULL STRENGTH OF CURRENT FROM CURRENT POLE RECORDS OF OBSERVING VESSEL ANCHORED AT MIDDLE OF COURSE.
DIRECTION OF CURRENT RELATIVE TO COURSE.
DIRECTION OF WIND RELATIVE TO COURSE AND FORCE BY BEAUFORT SCALE.
9A.M.
10-A.M.
11-A.M.
12-M
1-P.M.
2-P.M.
HIGH WATER
PROVINCETOWN
11h 43m A.M.
SCALE OF TIME IN HOURS

COMPARISON OF CURVE
SHOWING EFFECT OF CURRENT OBSER
BY DIFFERENT METHODS.


OBSERVATIONS TAKEN AT PROVINCETOWN, JUNE 28, 1905.
TRIAL OF THE U.S.S. CHARLESTON.
CURVE A, FROM SPEED & R.P.M.
CURVE B, FROM CURRENT POLE OBSERVATIONS
TIME OF HIGH WATER AT PROVINCETOWN, MASS. 7:48 A.M.


A, OBSERVATIONS DOUBTFUL
OBSERVATIONS TAKEN AT PROVINCETOWN, AUG. 16, 1905.
TRIAL OF THE U.S.S. PADUCAH.
CURVE B, FROM CURRENT POLE OBSER
CURVE A, FROM SPEED & R.P.M.


OBSERVATIONS TAKEN AT PROVINCETOWN, SEPT. 14, 1905.
TRIAL OF THE U.S.S. COLORADO.
TIME OF HIGH WATER AT PROVINCETOWN, MASS. 11:43 A.M.



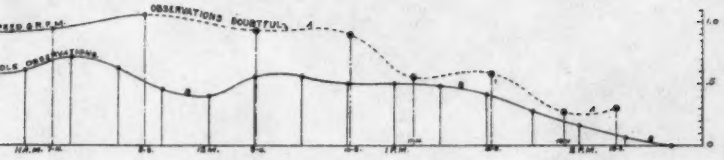
NOTE.

A = CURVE OF COMPONENTS OF CURRENT VELO
EFFECT OF WIND) IN THE DIRECTION OF COURSE,
OF SPEED AND R.P.M.

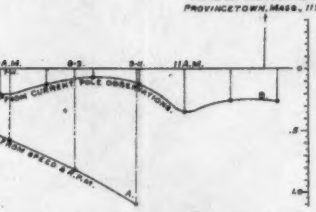
B = CURVE OF COMPONENTS OF CURRENT VELO
RECTION OF THE COURSE, TAKEN FROM CURREN
OF VESSEL ANCHORED AT MIDDLE OF COURSE.

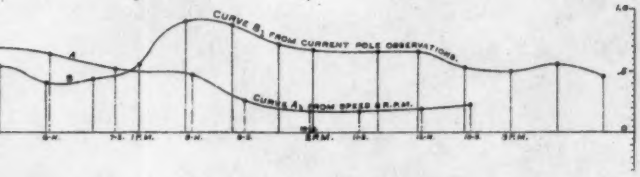
N OF CURVES
CURRENT OBSERVED
RENT METHODS.

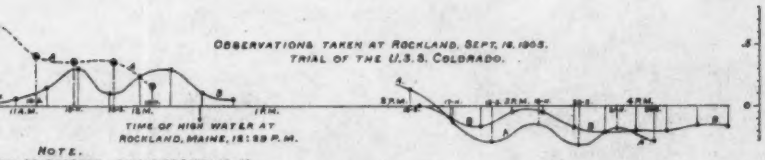
PLATE X.



TIME OF HIGH WATER AT
PROVINCETOWN, MASS., 11:58 A.M.







TIME OF HIGH WATER AT
ROCKLAND, MAINE, 12:39 P.M.

NOTE.

TS OF CURRENT VELOCITIES (INCLUDING
DIRECTION OF COURSE, TAKEN FROM CURVE

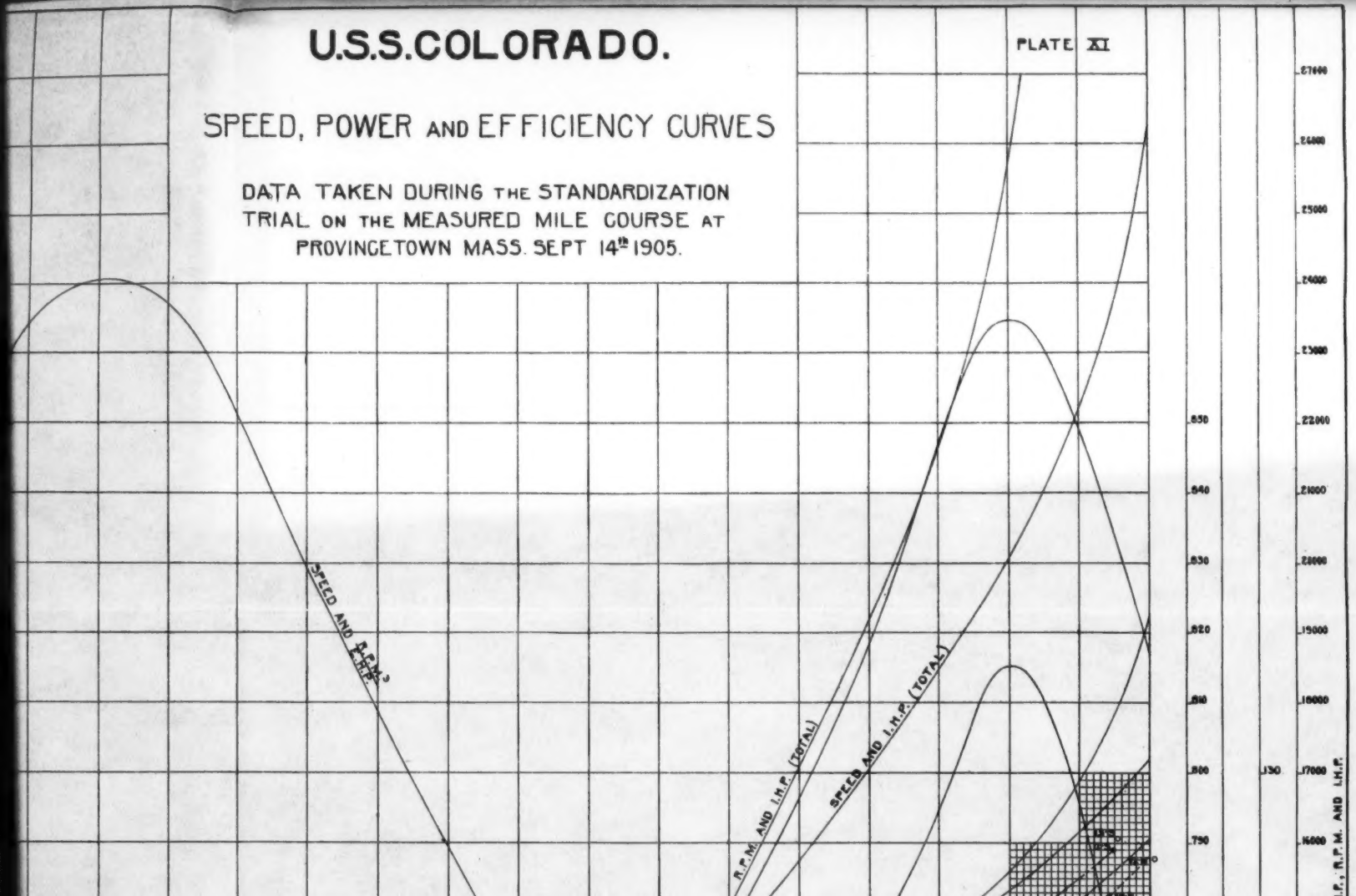
S OF CURRENT VELOCITIES IN THE BI-
, TAKEN FROM CURRENT POLE OBSERVATIONS
MIDDLE OF COURSE.

U.S.S.COLORADO.

PLATE XI

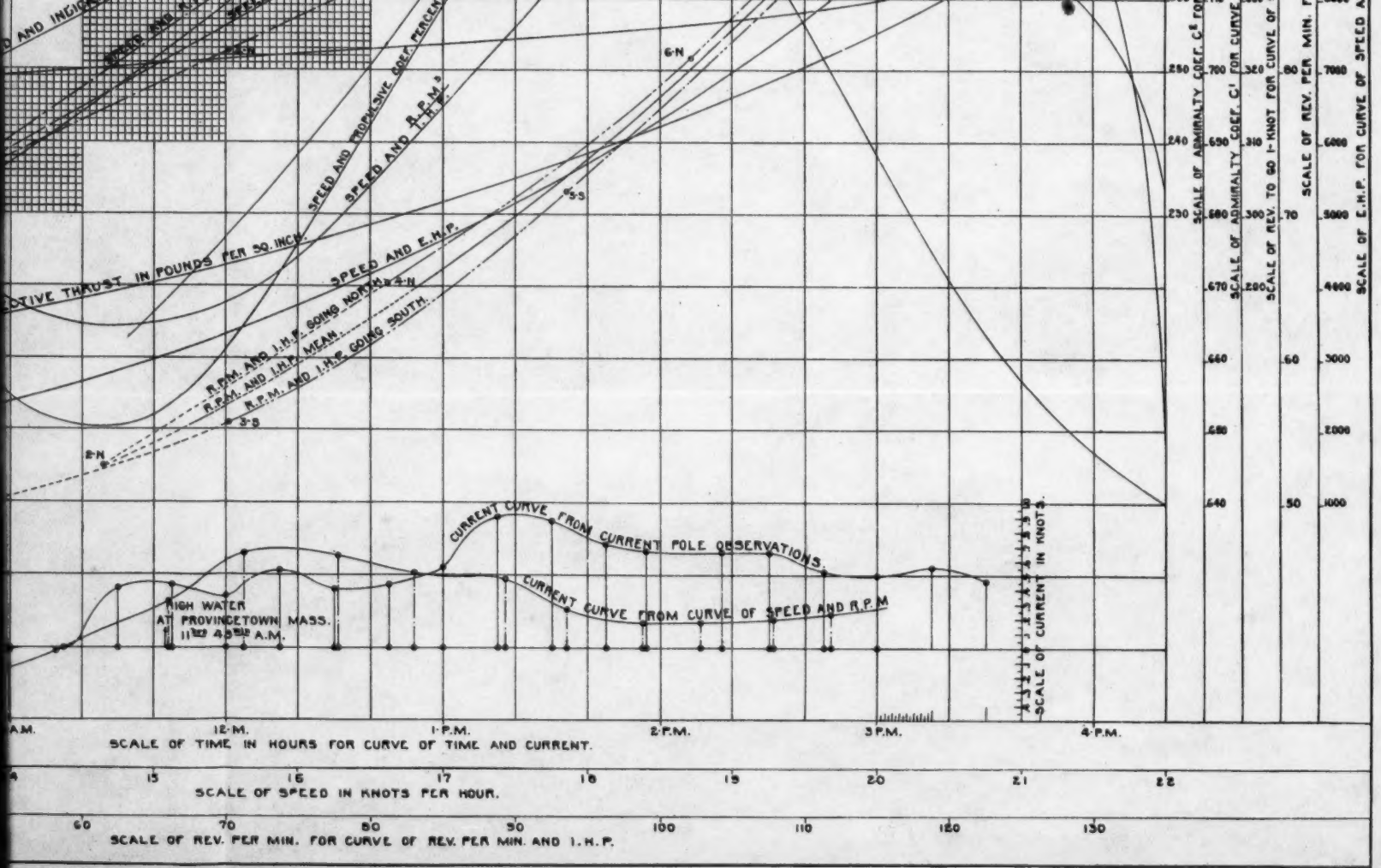
SPEED, POWER AND EFFICIENCY CURVES

DATA TAKEN DURING THE STANDARDIZATION TRIAL ON THE MEASURED MILE COURSE AT PROVINCETOWN MASS. SEPT 14th 1905.



SCALE OF ADMIRALTY COEF. C^2 FOR CURVE OF SPEED AND ADMIRALTY COEF. $C^2 = \frac{S^{2/3} DISP^{2/3}}{I.H.P.}$
SCALE OF ADMIRALTY COEF. C_1 FOR CURVE OF SPEED AND ADMIRALTY COEF. $C_1 = \frac{S^{2/3} AREA\ X^{2/3}}{I.H.P.}$
SCALE OF REV. TO GO 1-KNOT FOR CURVE OF SPEED AND REV. TO GO 1-KNOT.
SCALE OF REV. PER MIN. FOR CURVE OF SPEED AND REV. PER MIN.
SCALE OF I.H.P. FOR CURVES OF SPEED AND I.H.P., R.P.M. AND I.H.P.
SCALE OF E.H.P. FOR CURVE OF SPEED AND E.H.P.
SPEED AND R.P.M.
AND E.H.P.
SPEED AND PROPULSIVE COEF. PERCENT
AND INDICATED THRUST HORSE POWER
COEF. $C^2 = \frac{S^{2/3} DISP^{2/3}}{THP}$
AND SLIP PERCENT.
S.H.P.
R.P.M. AND I.H.P.
N-13-S
N-12-S
N-11-S
N-10-S
S-9
N-8
S-7
N-6
S-5

SCALE OF I.H.P. FOR CURVES OF SPEED AND I.H.P., R.P.M. AND I.H.P.
SCALE OF SPEED AND E.H.P.
R.P.M. AND I.H.P. (TOTAL)
SPEED AND I.H.P. (TOTAL)
SPEED AND R.P.M.
AND SLIP PERCENT.
AND INDICATED THRUST
N-8
N-9
N-10
S-6
S-5
11-S
13-S
N-8
N-9

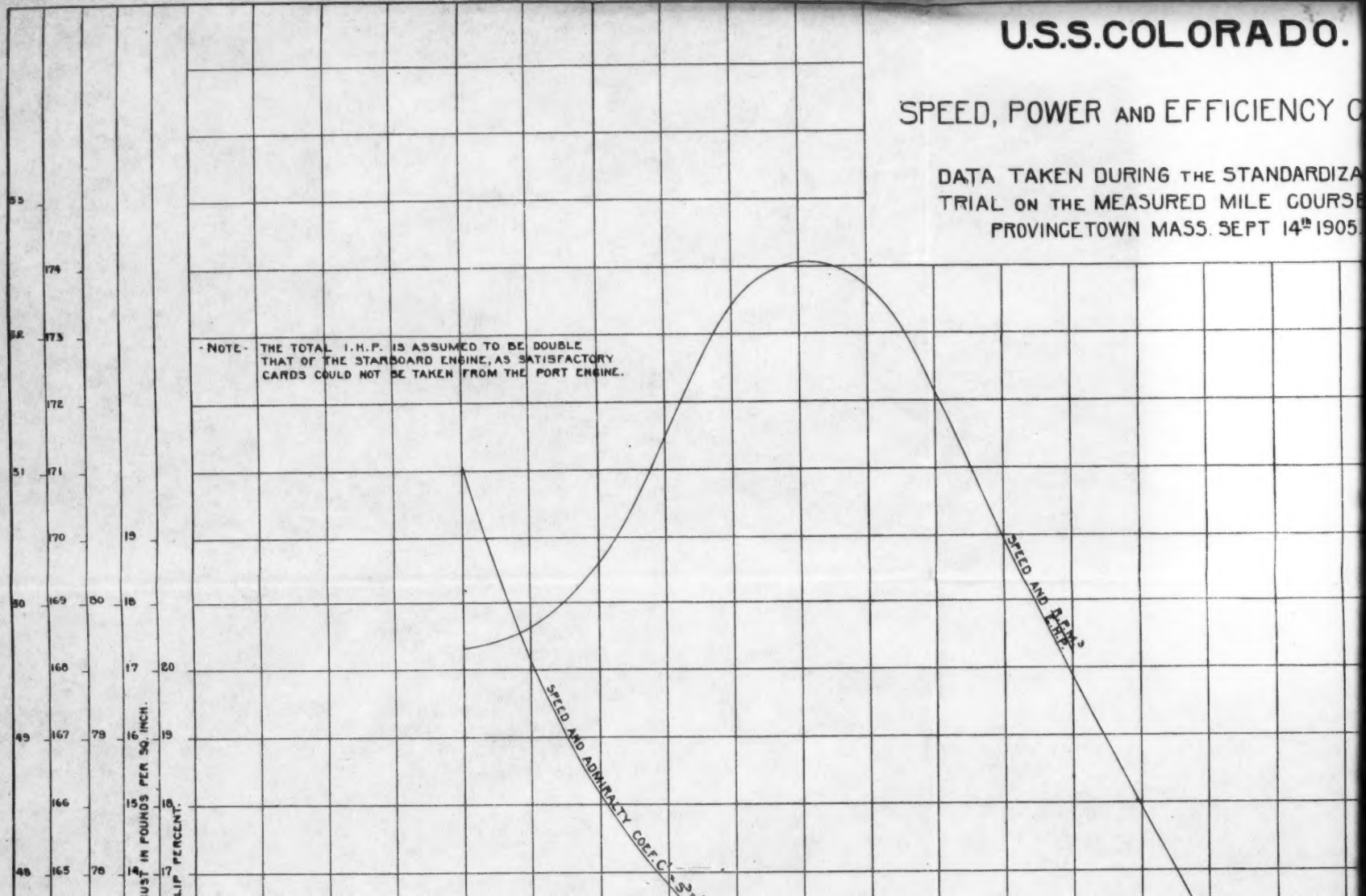

SPEED AND PROPULSIVE COEF. PERCENT
SPEED AND R.P.M.
SPEED AND I.H.P.
SPEED AND E.H.P.
R.P.M. AND I.H.P. GOING NORTH
R.P.M. AND I.H.P. MEAN
R.P.M. AND I.H.P. GOING SOUTH
6-N
5-S
4-N
3-S
2-N
CURRENT CURVE FROM CURRENT POLE OBSERVATIONS.
CURRENT CURVE FROM CURVE OF SPEED AND R.P.M
HIGH WATER AT PROVINCETOWN MASS. 11h 45m A.M.
SCALE OF CURRENT IN KNOTS.
SCALE OF ADMIRALTY COEF. C1 FOR CURVE
SCALE OF REV. TO GO 1-KNOT FOR CURVE OF
SCALE OF REV. PER MIN. F
SCALE OF E.H.P. FOR CURVE OF SPEED A
A.M.
12-M.
1-P.M.
2-P.M.
3 P.M.
4-P.M.
SCALE OF TIME IN HOURS FOR CURVE OF TIME AND CURRENT.
SCALE OF SPEED IN KNOTS PER HOUR.
SCALE OF REV. PER MIN. FOR CURVE OF REV. PER MIN. AND I.H.P.

U.S.S.COLORADO.

SPEED, POWER AND EFFICIENCY C

DATA TAKEN DURING THE STANDARDIZA
TRIAL ON THE MEASURED MILE COURSE
PROVINCETOWN MASS. SEPT 14th 1905.

- NOTE - THE TOTAL I.H.P. IS ASSUMED TO BE DOUBLE THAT OF THE STARBOARD ENGINE, AS SATISFACTORY CARDS COULD NOT BE TAKEN FROM THE PORT ENGINE.



SCALE OF SLIP PERCENT FOR CURVE OF SPEED AND SLIP PERCENT.

SCALE OF THRUST IN POUNDS PER SQ. INCH FOR CURVES OF SPEED AND INDICATED AND EFFECTIVE THRUST IN POUNDS

SCALE OF $\frac{R.P.M.}{I.H.P.^{\frac{1}{3}}}$ FOR CURVE OF SPEED AND $\frac{R.P.M.}{I.H.P.^{\frac{1}{3}}}$

SCALE OF $\frac{R.P.M.}{I.H.P.^{\frac{1}{3}}}$ FOR CURVE OF SPEED AND $\frac{R.P.M.}{I.H.P.^{\frac{1}{3}}}$

SCALE OF PROPULSIVE COEF. PERCENT FOR CURVE OF SPEED AND PROPULSIVE COEF. PERCENT.

SPEED AND REV. TO GO 1 KNOT.

SPEED AND ADMIRALTY COEF. $C^2 = \frac{S^3}{I.H.P.^{\frac{2}{3}}}$

SPEED AND SLIP PERCENT.

SPEED AND ADMIRALTY COEF. $C^2 = \frac{S^3 \times AREA\ P.\ SQ.\ FT.}{I.H.P.}$

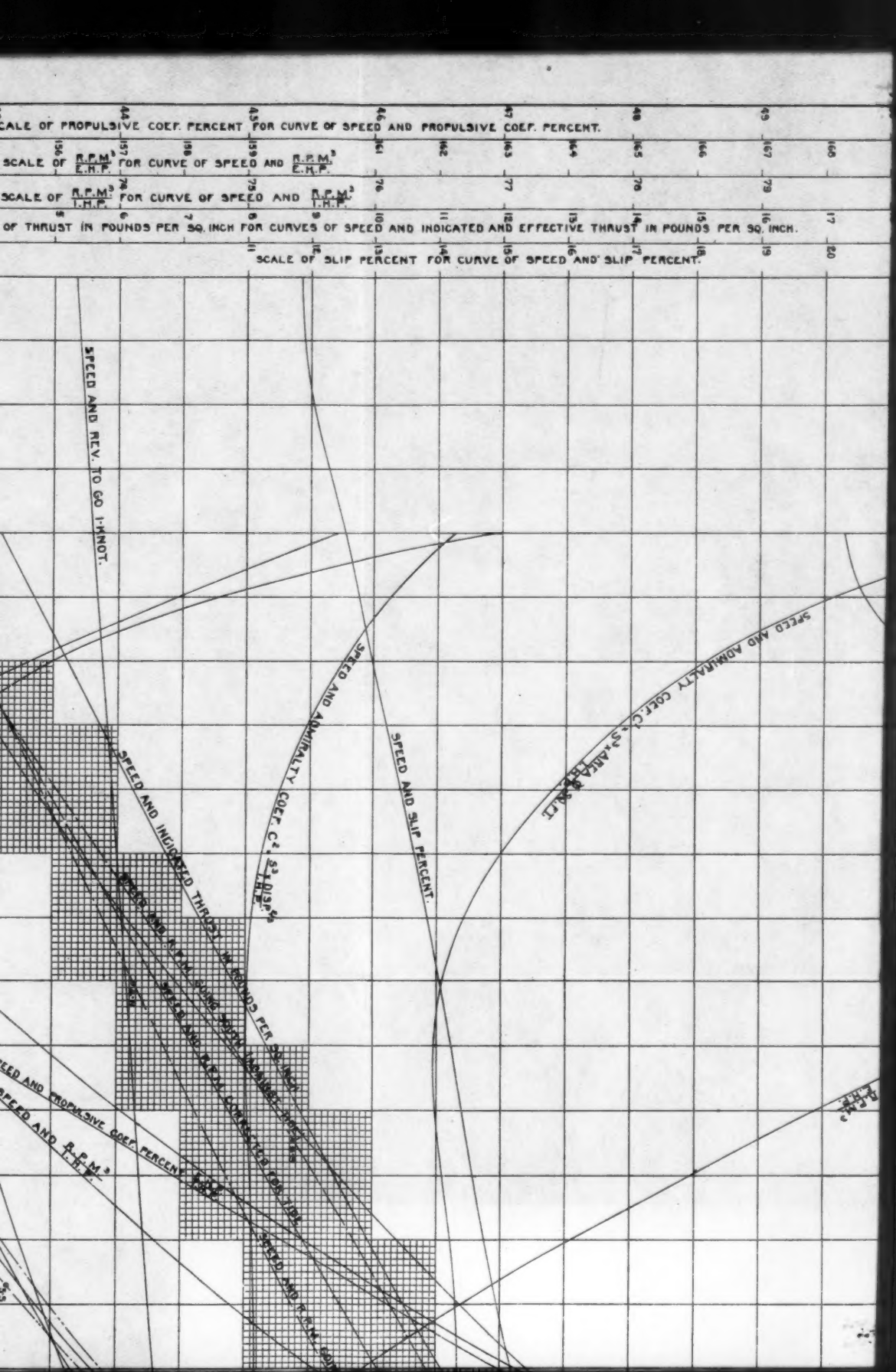
SPEED AND EFFECTIVE THRUST IN POUNDS PER SQ. INCH.

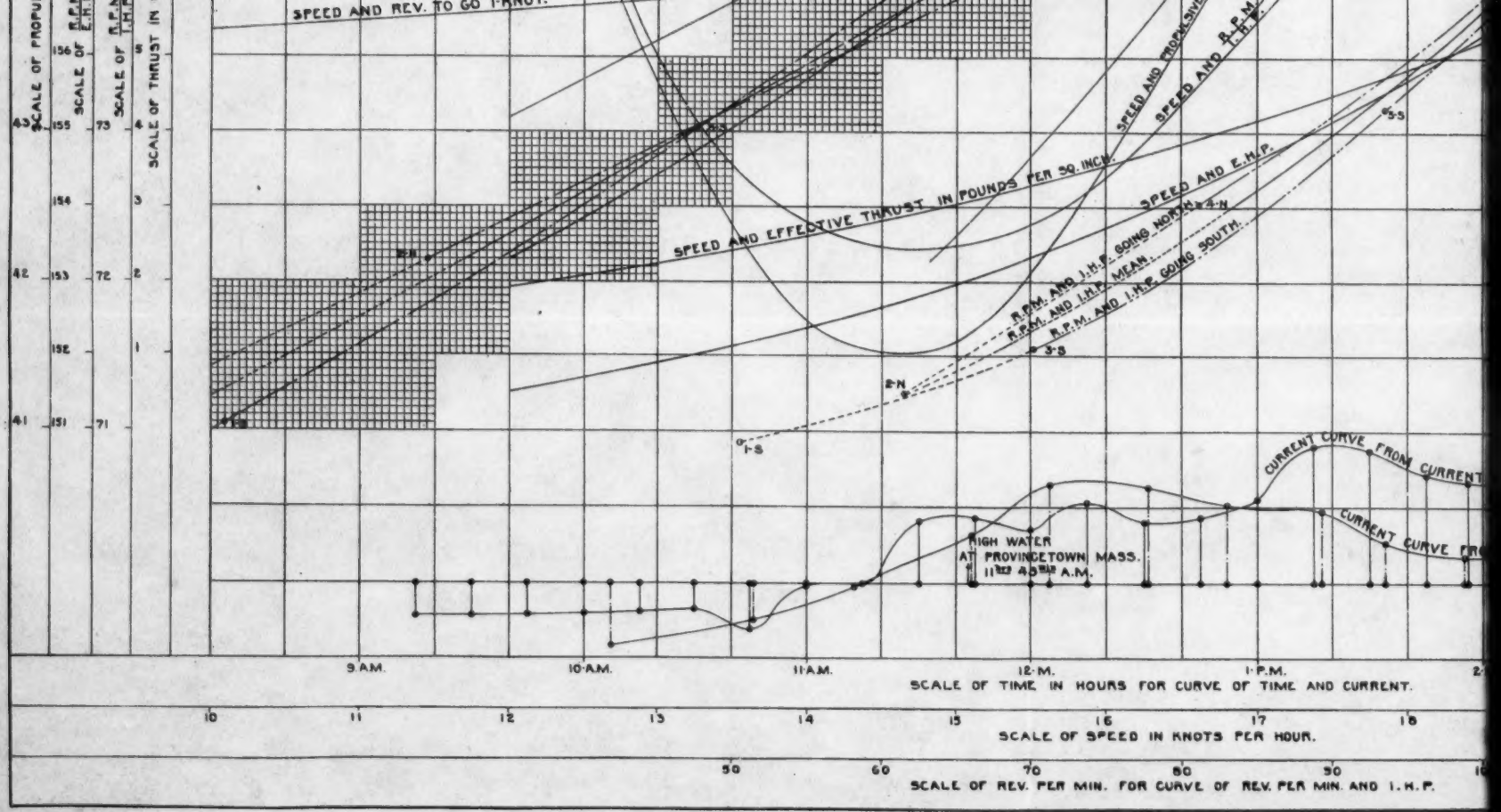
SPEED AND $\frac{R.P.M.}{I.H.P.^{\frac{1}{3}}}$

SPEED AND PROPULSIVE COEF. PERCENT

SPEED AND INDICATED THRUST IN POUNDS PER SQ. INCH

SPEED AND REV. TO GO 1 KNOT.


SCALE OF PROPULSIVE COEF. PERCENT FOR CURVE OF SPEED AND PROPULSIVE COEF. PERCENT.
SCALE OF R.P.M.³/E.H.P. FOR CURVE OF SPEED AND R.P.M.³/E.H.P.
SCALE OF R.P.M.³/I.H.P. FOR CURVE OF SPEED AND R.P.M.³/I.H.P.
OF THRUST IN POUNDS PER SQ. INCH FOR CURVES OF SPEED AND INDICATED AND EFFECTIVE THRUST IN POUNDS PER SQ. INCH.
SCALE OF SLIP PERCENT FOR CURVE OF SPEED AND SLIP PERCENT.
SPEED AND REV. TO GO 1 KNOT.
SPEED AND ADMIRALTY COEF. C¹ = 53.4 H.P.
SPEED AND ADMIRALTY COEF. C² = 53.4 H.P.
SPEED AND SLIP PERCENT.
SPEED AND ADMIRALTY COEF. C³ = 53.4 H.P.
SPEED AND INDICATED THRUST IN POUNDS PER SQ. INCH
SPEED AND R.P.M. SHOWING POINTS WHERE PROPELLER BEGINS TO RACE
SPEED AND REVS. CORRECTED FOR SLIP
SPEED AND REVS. CORRECTED FOR SLIP
SPEED AND PROPULSIVE COEF. PERCENT
SPEED AND R.P.M.³/I.H.P.


SCALE OF PROPULSIVE COEFFICIENT
SCALE OF R.P.M. AND I.H.P.
SCALE OF R.P.M. AND I.H.P.
SCALE OF THRUST IN POUNDS
SPEED AND REV. TO GO 1 KNOT.
SPEED AND PROPULSIVE COEFFICIENT.
SPEED AND R.P.M. AND I.H.P.
SPEED AND EFFECTIVE THRUST IN POUNDS PER SQ. INCH.
SPEED AND E.H.P.
R.P.M. AND I.H.P. GOING NORTH.
R.P.M. AND I.H.P. MEAN.
R.P.M. AND I.H.P. GOING SOUTH.
CURRENT CURVE FROM CURRENT
CURRENT CURVE FRO
HIGH WATER
AT PROVINCETOWN, MASS.
11H 45M A.M.
9 A.M.
10 A.M.
11 A.M.
12 M.
1 P.M.
SCALE OF TIME IN HOURS FOR CURVE OF TIME AND CURRENT.
SCALE OF SPEED IN KNOTS PER HOUR.
SCALE OF REV. PER MIN. FOR CURVE OF REV. PER MIN. AND I. H. P.

DATA TAKEN DURING STANDARDIZATION TRIAL OF THE U.S.S. COLORADO.
ON THE MEASURED MILE COURSE AT PROVINCETOWN, MASS., SEPT. 14, 1905.

[ObSE]RVED TIME [T]O 1-MILE. [ST]OP WATCHES. 2ND. AFTER OBS'VR. POTTS.	3RD. DAYTON	AVG. ½ SUM COLS. 5, 6 & 7.	TOTAL REVOLUTIONS TO GO 1-MILE. BY MECHANICAL COUNTERS ON DECK. ST'B'D.	PORT	MEAN ½ SUM COLS 9 & 10	DIF. COL 9 − COL 10	BY ELECTRIC PRINTING COUNTERS. STARBOARD ENGINE FOR'D. OBS'V'R	AFTER OBS'V'R	MEAN ½ SUM COLS 13 & 14	DIF. COL 13 − COL 14	PORT ENGINE. FOR'D. OBS'V'R.	AFTER OBS'V'R.	MEAN ½ SUM COLS 17 & 18	DIF. COL 17 − COL 18	DIF. COL 15 − COL 19	AVG. REV TO GO 1 M. ½ SUM COLS 15 & 19	REVOLUTIONS PER MINUTE OF ENGINES. BY MECHANICAL COUNTERS ON DECK. STAR'B'D FROM COLS. 7 & 9.	PORT FROM COLS. 7 & 10.	MEAN FROM COLS. 23 & 24	BY ELECTRIC PRINTING COUNTERS. STARBOARD ENG. FOR'D. OBS'V'R FROM COLS. 5 & 13.	AFTER OBS'V'R. FROM COLS. 6 & 14.	MEAN ½ SUM COLS. 26 & 27.	PORT ENGINE. FOR'D. OBS'V'R FROM COLS. 5 & 17.	AFTER OBS'V'R FROM COLS. 6 & 18.	MEAN ½ SUM COLS. 29 & 30	AVG. REV. BOTH ENGINES ½ SUM COLS. 28 & 31	SPEED CORR. TO AVG. OF OBSERVED TIMES.	NO. AND DIRECTION OF RUN.
6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34
M. S. 5-56.6	M. S. 5-56.4	M. S. 5-56.50	300.7	299.5	300.10	1.2	299.9	300.7	300.30	-.8	298.7	299.1	298.90	-.4	1.40	299.60	50.62	50.42	50.52	50.63	50.59	50.61	50.48	50.33	50.36	50.50	10.10	1-S.
5-14.2	5-14.1	5-14.15	322.30	323.1	322.70	-.8	322.9	322.3	322.60	.6	322.8	321.3	322.05	1.5	.55	322.32	61.57	61.72	61.64	61.52	61.55	61.54	61.51	61.36	61.43	61.49	11.46	2-N.
4-31.2	4-31.0	4-31.20	317.15	318.3	317.72	-1.15	317.0	317.3	317.15	-.3	317.2	317.0	317.10	.2	.05	317.13	70.22	70.47	70.35	70.13	70.20	70.16	70.18	70.13	70.16	70.16	13.27	3-S.
[3]-51.8	3-51.8	3-51.83	313.45	314.1	313.77	-.65	313.2	314.0	313.60	-.8	314.4	314.8	314.60	-.4	-1.0	314.10	81.13	81.30	81.21	81.03	81.28	81.16	81.35	81.48	81.41	81.29	15.53	4-N.
[3]-35.1	3-35.2	3-35.15	334.25	336.0	335.12	-1.75	334.8	335.0	334.90	-.2	336.6	337.0	336.80	-.4	-1.90	335.85	93.19	93.68	93.43	93.69	93.44	93.51	94.20	94.00	94.10	93.81	16.73	5-S.
[3]-5.2	3-5.0	3-5.10	314.50				314.2	314.6	314.40	-.4	316.2	316.5	316.35	-.3	-1.95	315.37	102.00			102.23	101.92	102.08	102.89	102.54	102.71	102.40	18.45	6-N.
[3]-7.0	3-6.8	3-6.83	337.45	339.0	338.22	-1.55	336.8	338.0	337.40	-1.2	337.6	338.6	338.10	-1.0	-.70	337.75	108.39	108.89	108.64	108.24	108.45	108.34	108.49	108.64	108.57	108.46	19.27	7-S.
[2]-56.4	2-56.6	2-56.50	324.65	319.6	322.12	5.05	324.1	323.8	323.95	.3	320.1	320.1	320.10	0.0	3.85	322.02	110.30	108.58	109.58	110.18	110.14	110.16	108.82	108.88	108.85	109.50	20.40	8-N.
[2]-53.2	2-53.0	2-53.10	338.95	340.2	339.57	-1.25	336.2	336.6	336.40	-.4	340.4	341.1	340.75	-.7	-4.35	338.57	117.56	117.99	117.77	116.47	116.60	116.54	117.92	118.16	118.04	117.29	20.80	9-S.
[2]-45.9	2-46.4	2-45.90	334.35				336.3	336.0	336.15	.3	334.2	334.1	334.15	.1	2.00	335.15	120.78			121.63	121.52	121.57	120.87	120.83	120.85	121.21	21.70	10-N.
[2]-45.8	2-45.8	2-45.63	343.95	343.0	343.47	.95	344.0	344.5	344.25	-.5	342.6	343.3	342.95	-.7	1.30	343.60	124.47	124.13	124.30	124.71	124.67	124.69	124.21	124.23	124.22	124.46	21.74	11-S.
2-46.8	2-43.4	2-43.40	337.65	334.9	336.27	2.75	339.0	337.8	338.40	1.2	335.7	334.6	335.15	1.1	3.25	336.77	123.98	122.97	123.48	124.48	124.50	124.49	123.27	123.32	123.27	123.88	22.08	12-N.
[2]-45.6	2-45.4	2-45.43	345.00	346.3	345.65	-1.30	344.3	344.7	344.50	-.4	346.3	346.4	346.35	-.1	-1.85	345.42	125.15	125.62	125.38	124.97	124.89	124.93	125.70	125.51	125.61	125.27	21.77	13-S.

…NING AVERAGE TIMES.

…OF FINAL FOUR HOUR TRIAL.
…ING TRIAL (BY PATENT LOG)..... 21.62 KNOTS.
…G TRIAL (ALL MACHINERY)........ 24091.7
…TING SURFACE..................... .3515
…RATE SURFACE..................... 15.0673
…SQ. FT. OF GRATE SURFACE PER HR..... 31.62 LBS.
… I.H.P. PER HOUR.................. 2.164 "
…R DURING TRIAL
…PTION PER HOUR.................. 52625 LBS.
…T................................ 24 FT. 9 INS.
…ENT.............................. 14240 TONS.

…Y ENGINEERS DURING STANDARDIZATION TRIAL.

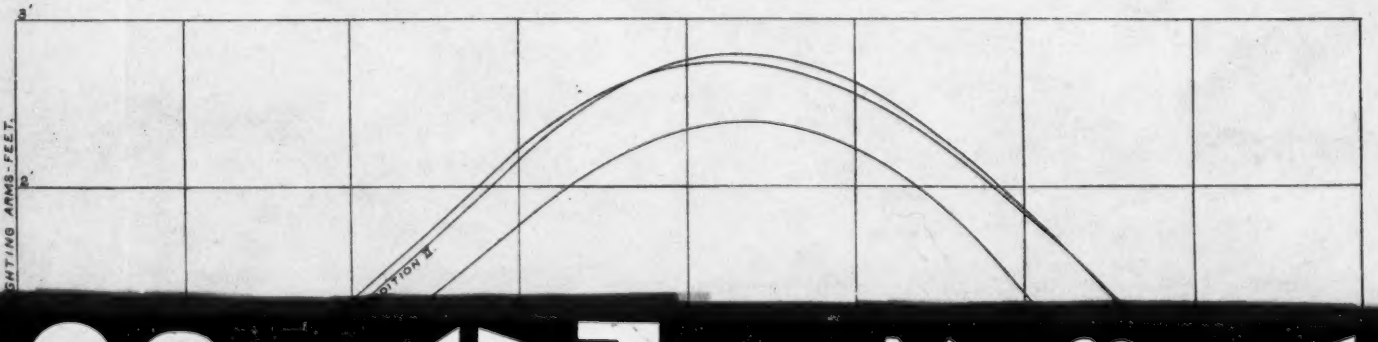
…S.	REVOLUTIONS OR DOUBLE STROKES.	…TER. IN…	INDICATED HORSE POWER. STARBOARD ENGINE	PORT ENGINE	TOTAL	AUXILIARIES.	AUXS.	TOTAL	TOTAL	…UN.

Y ENGINEERS DURING STANDARDIZATION TRIAL.

S.		REVOLUTIONS OR DOUBLE STROKES.								AIR PRESSURE IN INCHES OF WATER.	INDICATED HORSE POWER.																													NUMBER AND DIRECTION OF RUN.
											STARBOARD ENGINE.					PORT ENGINE.					TOTAL OF BOTH MAIN ENG'S. SUM COLS. 72 & 77	AUXILIARIES.														AUX'S. CONT'G TO PROP. SUM COLS. 81 TO 88 INC.	TOTAL MAIN ENGS. & ALL AUX'S. SUM COLS. 78 & 89	TOTAL MAIN ENGS. & AUX'S. CONTRIB-UTING TO PRO-PULSION SUM COLS 78 & 90.	
	FEED.	CIRCULATING PUMP.	MAIN FEED PUMP.	AUX. FEED PUMP.	HOTWELL PUMP.	WATER SERVICE PUMP.	BILGE PUMP.	AUX. CONDENSER PUMP.	FIREROOM BLOWERS.		H.P. CYLINDERS.	I.P. CYLINDERS.	FORWARD L.P. CYLINDERS.	AFTER L.P. CYLINDERS.	TOTAL SUM COLS. 68, 69, 70 & 71.	H.P. CYLINDERS.	L.P. CYLINDERS.	FORWARD L.P. CYLINDERS.	AFTER L.P. CYLINDERS.	TOTAL SUM COLS. 73, 74, 75 & 76		DYNAMO	STEERING ENGINE.	MAIN CIRCULATING PUMP ENG.	MAIN FEED PUMPS.	AUX. FEED PUMPS.	HOTWELL PUMPS	WATER SERVICE PUMPS.	BILGE PUMPS.	AUX. CONDR. PUMPS.	FORCED DFT. BLOWERS.	TOTAL ALL AUX'S. SUM COLS. 79 TO 88 INC.				
57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93
											942.56	222.92	149.56	136.45	851.49	209?	230	165	172	776?																		1-S.
											543.9	438.37	253.27	271.13	1506.67	254?	515	226	255	1250?																		2-N.
											736.1	670.6	334.5	357.7	2098.9	368.?	480.8?	363.5	341.7	1554?																		3-S.
											1659.8	1144.6	572.49	665.1	4041.99	1862.1	1163.8	574.3	578.6	4178.8																		4-N.
											1787.4	1740.7	873.65	893.06	5304.8	1807.6	1722	903	912.7	5345.3																		5-S.
											2314.7	2343.8	1250.2	1250.2	7158.9	2329	2272	1194	1173	6968																		6-N.
											2433.5	2292.7	1680.6	1630.1	8036.9	2627	2381	1758	1617	8383						NO DATA										7-S
											2868.7	2898.7	1438.5	1461.4	8789.3	2618	2819	1343	1281	8066																		8-N.
											3241.0	3380.2	1640.1	1549.3	9810.6	2973	3334	1631	1594	9532																		9-S.
											3917.3	3663.6	1830.0	1874.1	11285		4222.8	1254?	1756.6																			10-N
											3984.6	3989.5	2455.7	2313.6	12753.3	CARDS UNRELIABLE.																							11-S.
											4156.4	4358.6	2030.6	1972.6	12518.2			1277?	1871																			12-N
											4070.9	4520.2	2104.2	2052.4	12747.7			1659?	2382																			13-S.

CURVES OF STATICAL STABILITY.

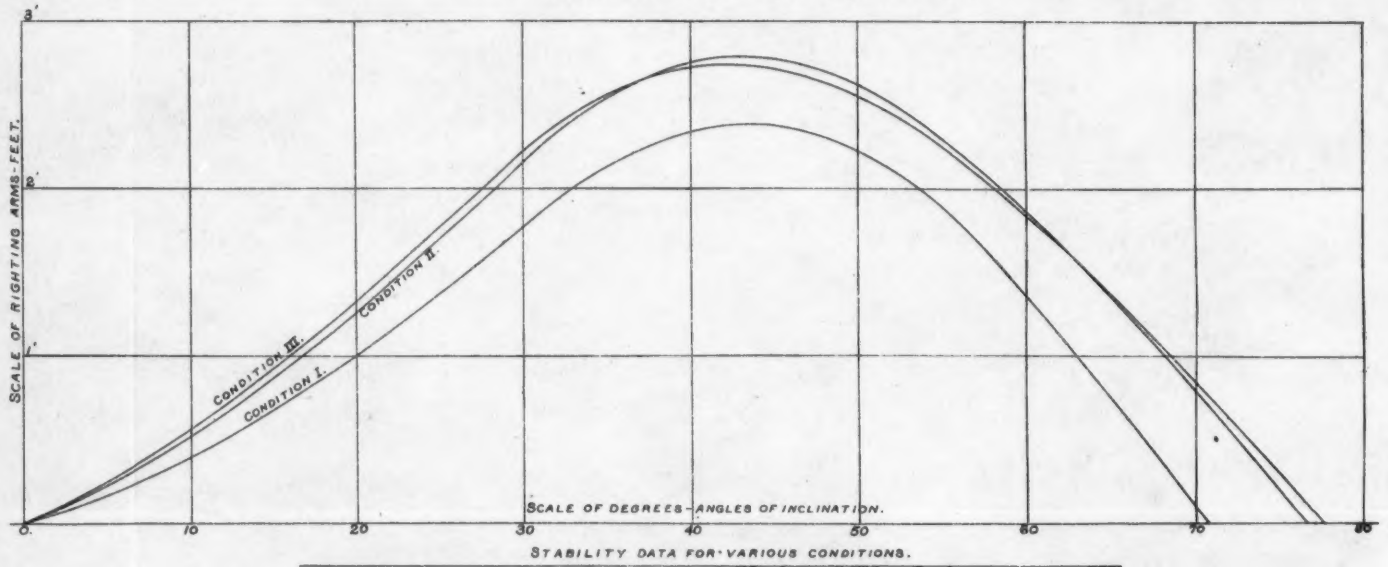
ADMIRALTY COEF. C_2. $\frac{V^3 \times D.^{\frac{2}{3}}}{I.N.P.}$
104
302.8
285.1



HOTWEL	FEED.	CIRCULATI PUMP.	MAIN FE PUMP.	AUX. FEE PUMP.	HOTWEL PUMP.	WATER BER PUMP.	BILGE PUMP.	AUX. CONDE PUMP.	FIRE RO BLOWE	AIR PRESS INCHES OF	H.P. CYLINDER	I.P. CYLINDER	FORWARD L.P. CYLINDER	AFTER L.P. CYLINDER	SUM COLS. 68,69, 70 & 71.	H.P. CYLINDER	L.P. CYLINDER	FORWARD L.P. CYLINDER	AFTER L.P. CYLINDER	SUM COLS. 73,74, 75 & 76	MAIN ENG'S. SUM COLS. 72 & 77	DYN.A	STEERI ENGINE	MAIN CI CULATIN PUMP EA	MAIN FE PUMPS	AUX. FE PUMPS	HOTWE PUMPS	WATE SERVIC PUMPS	BILGE PUMPS	AUX. CON PUMPS	FORCED BLOWE	AUX'S. SUM COLS. 70 TO 88 INC.	PROP. SUM COLS. 81 TO 88 INC.	AUX'S. SUM COLS. 78 & 89	UTING TO PRO-PULSION SUM COLS 78 & 90.	NUME DIRECTI
57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93
											342.55	222.92	149.56	136.45	851.48	209?	230	165	172	776?																1-3.
											543.9	436.37	253.27	271.13	1506.67	254?	515	226	255	1250?																2-W.
											736.1	670.6	334.5	357.7	2098.9	368.?	480.8?	363.5	341.7	1554?																3-S.
											1659.8	1144.6	572.49	665.1	4041.99	1862.1	1163.8	574.3	578.6	4178.8																4-N.
											1787.4	1740.7	873.65	883.06	5304.8	1807.6	1722	903	912.7	5345.3																5-S.
											2314.7	2343.8	1250.2	1250.2	7158.9	2329	2272	1194	1173	6968																6-N.
											2433.5	2292.7	1680.6	1630.1	8036.9	2627	2381	1758	1617	8383	———	———	———	———	———	———	———	———	———	———	NO DATA				7-S
											2869.7	2999.7	1438.5	1461.4	8769.3	2618	2819	1343	1281	8066																8-N.
											3241.0	3380.2	1640.1	1549.3	9810.6	2973	3334	1631	1594	9532																9-S.
											3917.3	3683.6	1830.0	1874.1	11285			4222.8	1254?	1756.6																10-N
											3994.6	3989.5	2455.7	2313.6	12753.3	CARDS UNRELIABLE.																					11-S.
											4156.4	4358.6	2030.6	1972.6	12518.2			1277?	1871																	12-N
											4070.2	4520.2	2104.2	2052.4	12747.7			1659?	2382																	13-S.

ADMIRALTY COEF. C_2. $\frac{V^2 \times D.^{\frac{2}{3}}}{I.H.P.}$
104
302.8
285.1
276.7
271.7
270.0
270.5
271.2
271.2
269.4
265.1
233.6

CURVES OF STATICAL STABILITY.

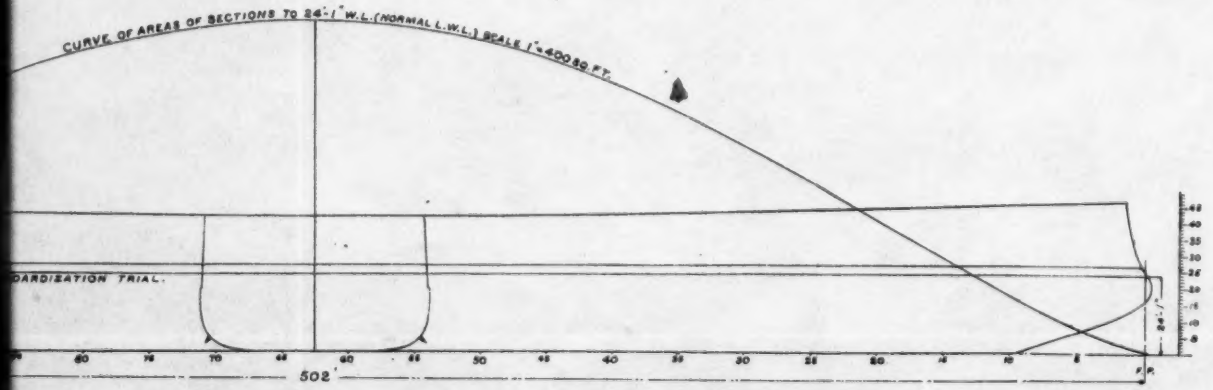


STABILITY DATA FOR VARIOUS CONDITIONS.

	271.2
7	271.2
7	269.4
	265.1
4	233.6
TION.	

STABILITY DATA FOR VARIOUS CONDITIONS.

CONDITIONS.	MAX. RIGHTING ARM	ANG. AT WHICH MAX. RIGHTING ARM OCCURS.	RANGE OF STABILITY.	METACENTRIC HEIGHT.
I. SHIP LIGHT. NO CREW, COAL, AMMN. STORES OR FRESH WATER ON BOARD.	2.38 FT.	43°- 8'	70°- 38'	2.00 FT.
II. SHIP IN NORMAL CONDITION. TRIAL ALLOWANCE OF COAL, STORES &C. ON BOARD.	2.78 "	42°- 55'	76°- 41'	2.66 "
III. SHIP FULLY LOADED.	2.74 "	42°- 30'	77°- 30'	2.52 "



U. S. S. COLORADO.

DATA TAKEN DURING THE PRELIMINARY AND FINAL ACCEPTANCE TRIALS.

PRINCIPAL DIMENSIONS.

HULL.

LENGTH BETWEEN PERPENDICULARS	502'-0"
LENGTH OVER ALL	504'-0½"
BEAM AT NORMAL L.W.L.	69'-1"
BEAM EXTREME	69'-6½"
DRAFT, MEAN ..... (STANDARDIZATION TRIAL)	24'-1"
DISPLACEMENT	13780 TONS.
WETTED SURFACE	43011
AREA OF MIDSHIP SECTION	1585.5 SQ.FT.
AREA OF L.W.L. PLANE	23840
TONS PER INCH IMMERSION	57.00
MOMENT TO PRODUCE 1" OF HEEL	639.6 FT. TONS.
COEF. OF FINENESS, BLOCK	.5765
COEF. OF FINENESS, MIDSHIP SEC.	.9574
COEF. OF FINENESS, NORMAL L.W.L.	.6903
CYLINDRICAL COEFFICIENT	.6022
KEEL PROJECTS BELOW SKIN DRAFT	1 3/16

ENGINES.

TYPE	TRIPLE-EXPANSION, VERTICAL-INVERTED DIRECT-ACTING.
NUMBER	2
CYLINDERS	4
HIGH PRESSURE DIAM.	38½ INS.
INTERMEDIATE PRESSURE, DIAM.	63½ "
FORWARD LOW PRESSURE, DIAM.	74 "
AFTER LOW PRESSURE, DIAM.	74 "
STROKE	4 FT.
I.H.P. (MAXIMUM) ALL MACHINERY	

BOILERS.

TYPE	NICLAUSSE IMPROVED.
NUMBER	32
GRATE SURFACE	1600 SQ. FT.
HEATING SURFACE	68537 " "

PROPELLERS.

TYPE	THREE BLADED.
NUMBER	2
MATERIAL	MANGANESE BRONZE
DIAMETER	18 FT.
MEAN PITCH, ON TRIAL	22 "
HELICOIDAL AREA, TOTAL ALL PROPELLERS	184 SQ. FT.
PROJECTED AREA, TOTAL ALL PROPELLERS	148 " "

DATA TAKEN DURING STANDARDIZATION T
ON THE MEASURED MILE COURSE AT PR

No. AND DIRECTION OF RUN	TIME OF BEGINNING EACH RUN.	TIME MIDDLE OF RUN COL 2 + ½ COL. 8.	TIME BET. MIDDLE OF RUNS FROM COL. 3.	OBSERVED TIME TO GO 1-MILE BY STOP WATCHES 1ST. FOR'D. OBS'VR. WOODWARD	2ND. AFTER OBS'VR. POTTS.	3RD. DAYTON	AVG. ⅓ SUM COLS. 5, 6 & 7.	TOTAL REVOLUTIONS TO GO 1-MILE BY MECHANICAL COUNTERS ON DECK. STB'D.	PORT	MEAN ½ SUM COLS 9 & 10	DIF. COL 9 - COL 10	BY ELECTRIC PRINTING COUNTE STARBOARD ENGINE FOR'D. OBS'VR.	AFTER OBS'VR.	MEAN ½ SUM COLS 13 & 14	DIF. COL 13 - COL 14	PORT ENGINE. FOR'D. OBS'VR.	AFTER OBS'VR.	MEAN ½ SUM COLS 17 & 18
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
1-S.	H. M. 10-7	H. M. S. 10-9-51.25		M. S. 5-35.4	M. S. 5-36.6	M. S. 5-36.4	M. S. 5-36.50	300.7	299.5	300.10	1.2	299.9	300.7	300.30	-.8	298.7	299.1	298.90
2-N.	10-46	10-48-37.07	M. S. 38-45.82	5-14.9	5-14.2	5-14.1	5-14.15	322.30	323.1	322.70	-.8	322.9	322.3	322.60	.6	322.8	321.3	322.05
3-S.	11-13	11-15-15.6	26-38.53	4-31.2	4-31.2	4-31.0	4-31.20	317.15	318.3	317.72	-1.15	317.0	317.3	317.15	-.3	317.2	317.0	317.10
4-N.	11-44	11-45-55.9	30-40.30	3-51.9	3-51.8	3-51.8	3-51.83	313.45	314.1	313.77	-.65	313.2	314.0	313.60	-.8	314.4	314.8	314.60
5-S.	12-5	12-6-47.52	20-51.61	3-34.4	3-35.1	3-35.2	3-35.15	334.25	336.0	335.12	-1.75	334.8	335.0	334.90	-.2	336.6	337.0	336.80
6-N.	12-31	12-32-32.5	25-44.98	3-4.4	3-5.2	3-5.0	3-5.10	314.50				314.2	314.6	314.40	-.4	316.2	316.5	316.35
7-S.	12-52	12-53-39.4	21-00.9	3-6.7	3-7.0	3-6.8	3-6.83	337.45	338.0	337.72	-1.55	336.8	338.0	337.40	-1.2	337.6	338.6	338.10
8-N.	1-17	1-18-26.3	24-54.9	2-56.5	2-56.4	2-56.6	2-56.50	324.65	319.6	322.12	5.05	324.1	323.8	323.95	.3	320.1	320.1	320.10
9-S.	1-34	1-35-26.5	16-58.2	2-53.2	2-53.2	2-53.0	2-53.10	338.95	340.2	339.57	-1.25	336.2	336.6	336.40	-.4	340.4	341.1	340.75
10-N.	1-55	1-56-22.95	20-56.45	2-45.9	2-45.9	2-44.4	2-45.90	334.95				336.3	336.0	336.15	.3	334.2	334.1	334.15
11-S.	2-11	2-12-22.9	16-59.95	2-45.5	2-45.8	2-45.8	2-45.63	343.95	343.0	343.47	.95	344.0	344.8	344.25	-.5	342.6	343.3	342.95
12-N.	2-31	2-32-21.7	19-58.8	2-43.4	2-43.8	2-43.4	2-43.40	337.65	334.9	336.27	2.75	339.0	337.6	338.40	1.2	335.7	334.6	335.15
13-S.	2-47	2-48-23.7	16-01.0	2-45.3	2-45.6	2-45.4	2-45.43	345.00	346.3	345.65	-1.30	344.3	344.7	344.50	-.4	346.3	346.4	346.35

⊠ FIGURES NOT USED IN OBTAINING AVERAGE TIMES.

PERFORMANCE ON PRELIMINARY SPEED TRIAL.

AVERAGE SPEED DURING TRIAL	22.244 KNOTS.
AVERAGE I.H.P. DURING TRIAL (ALL MACHINERY)	27309.33
I.H.P. PER SQ.FT. OF HEATING SURFACE	.3986
I.H.P. PER SQ.FT. OF GRATE SURFACE	17.07
COAL CONSUMED PER SQ.FT. OF GRATE SURFACE PER HR.	
COAL CONSUMED PER I.H.P. PER HOUR	
TOTAL LOSS OF WATER DURING TRIAL	
TOTAL COAL CONSUMPTION PER HOUR	
AVERAGE MEAN DRAFT	24FT. 1⅝ INS.
AVERAGE DISPLACEMENT	13823 TONS.

PERFORMANCE OF FINAL FOUR HOUR TRIAL.

AVERAGE SPEED DURING TRIAL (BY PATENT LOG)	21.62 KNOTS.
AVERAGE I.H.P. DURING TRIAL (ALL MACHINERY)	24091.7
I.H.P. PER SQ.FT. OF HEATING SURFACE	.3515
I.H.P. PER SQ.FT. OF GRATE SURFACE	15.0673
COAL CONSUMED PER SQ.FT. OF GRATE SURFACE PER HR.	31.62 LBS.
COAL CONSUMED PER I.H.P. PER HOUR	2.184 "
TOTAL LOSS OF WATER	
TOTAL COAL CONSUMPTION PER HOUR	52625 LBS.
AVERAGE MEAN DRAFT	24FT. 9INS.
AVERAGE DISPLACEMENT	14240 TONS.

DATA TAKEN BY ENGINEERS DURING STANDARDIZATION

NUMBER AND DIRECTION OF RUN.	TIME OF BEGINNING EACH RUN.	STEAM ENGINE.	STEAM PRESSURE. BOILERS ABOVE ATMOSPHERE.	STEAM CHEST. ABSOLUTE.	RECEIVER ABSOLUTE.	RECEIVER ABSOLUTE.	MEAN EFFEC. PRESSURE. F'WD. CYLINDER.	F'WD. CYLINDER.	AFTER L.P. CYLINDER.	AFTER L.P. CYLINDER.	REDUCED TO ATMOSPHERIC PRESSURE.	VACUUM IN INCHES.	CUT OFF IN CYLS. FROM BEGINNING. F'WD.	AFTER	TEMPERATURES. OUTSIDE AIR.	ENGINE ROOM WORKING PLATFORM.	BOILER ROOM WORKING PLATFORM.	INJECTION.	DISCHARGE.	WELL.	COOLED.	REVOLUTIONS OR DOUBLE STROKES. CIRCULATING PUMP.	MAIN FEED PUMP.	AUX. FEED PUMP.	AIR PUMP.	WELL PUMP.	SERVICE PUMP.	BILGE PUMP.	CONDENSER PUMP.	ENGINE ROOM BLOWERS.	PRESSURE IN ATM. OF WATER.

	STARBOARD ENGINE.				PORT ENGINE.			
				TOTAL SUM				

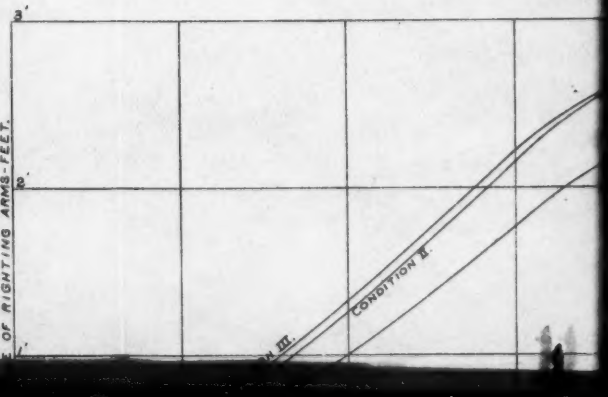
DATA TAKEN BY ENGINEERS DURING STANDARDIZATION

NUMBER AND DIRECTION OF RUN.	WHICH ENGINE.	STEAM PRESSURE. IN BOILER ABOVE ATMOSPHERE.	STEAM PRESSURE. AT END ABOVE ATMOSPHERE	STEAM PRESSURE. IN ST. RECEIVER ABSOLUTE.	STEAM PRESSURE. IN ST. RECEIVER ABSOLUTE.	STEAM PRESSURE. IN 2D. RECEIVER ABSOLUTE.	MEAN EFFEC. PRESSURE. H.P. CYLINDER.	MEAN EFFEC. PRESSURE. I.P. CYLINDER.	MEAN EFFEC. PRESSURE. FORWARD L.P. CYLINDER.	MEAN EFFEC. PRESSURE. AFTER L.P. CYLINDER.	MEAN EFFEC. PRESSURE. REFERRED TO LOW PRESSURE.	VACUUM IN INCHES.	CUT OFF IN CYLS. FROM BEGINNING. H.P.	CUT OFF IN CYLS. FROM BEGINNING. I.P.	CUT OFF IN CYLS. FROM BEGINNING. FORWARD L.P.	CUT OFF IN CYLS. FROM BEGINNING. AFTER L.P.	TEMPERATURES. OUTSIDE AIR.	TEMPERATURES. ENGINE ROOM WORKING PLATFORM.	TEMPERATURES. FIREROOM WORKING LEVEL.	TEMPERATURES. INJECTION.	TEMPERATURES. DISCHARGE.	TEMPERATURES. HOTWELL.	TEMPERATURES. FEED.	REVOLUTIONS OR DOUBLE STROKES. CIRCULATING PUMP.	REVOLUTIONS OR DOUBLE STROKES. MAIN FEED PUMP.	REVOLUTIONS OR DOUBLE STROKES. AUX. FEED PUMP.	REVOLUTIONS OR DOUBLE STROKES. HOTWELL PUMP.	REVOLUTIONS OR DOUBLE STROKES. WATER SERVICE PUMP.	REVOLUTIONS OR DOUBLE STROKES. BILGE PUMP.	REVOLUTIONS OR DOUBLE STROKES. AUX. CONDENSER PUMP.	REVOLUTIONS OR DOUBLE STROKES. FIREROOM BLOWERS.	AIR PRESSURE IN INCHES OF WATER.	INDICATED HORSE POWER. STARBOARD ENGINE. H.P. CYLINDERS.	I.P. CYLINDERS.	FORWARD L.P. CYLINDERS.	AFTER L.P. CYLINDERS.	TOTAL SUM COLS. 68, 69, 70 & 71.	PORT ENGINE. H.P. CYLINDERS.	[illegible]
35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	
1-S.	ST'B'D. PORT																																342.55	222.92	149.56	136.45	851.48	209?	
2-N.	ST'B'D. PORT																																543.9	438.37	253.27	271.13	1506.67	254?	
3-S.	ST'B'D. PORT																																736.1	670.6	334.5	357.7	2098.9	368.?	
4-N.	ST'B'D. PORT																																1659.8	1144.6	572.49	685.1	4041.99	1862.1	
5-S.	ST'B'D. PORT																																1797.4	1740.7	873.66	893.06	5304.8	1807.6	
6-N.	ST'B'D. PORT																																2314.7	2343.8	1250.2	1250.8	7158.3	2329	
7-S.	ST'B'D. PORT													NO DATA.																			2433.5	2292.7	1660.6	1630.1	8036.9	2627	
8-N.	ST'B'D. PORT																																2868.7	2939.7	1438.5	1461.4	8769.3	2618	
9-S.	ST'B'D. PORT																																3241.0	3380.2	1640.1	1549.3	9810.6	2973	
10-N.	ST'B'D. PORT																																3817.3	3663.6	1830.0	1874.1	11285		
11-S.	ST'B'D. PORT																																3994.5	3989.5	2455.7	2313.6	12753.3	CARD	
12-N.	ST'B'D. PORT																																4156.4	4358.6	2030.6	1972.6	12518.2		
13-S.	ST'B'D. PORT																																4070.9	4520.2	2104.2	2052.4	12747.7		

CURV

RESULTS OF CALCULATIONS FOR EFFICIENCY CURVES.

TRUE SPEED	TRUE I.H.P.	E.H.P.	PROPULSIVE COEF. E.H.P. / I.H.P.	REV. PER. MIN.³ / I.H.P.	REV. PER. MIN³ / E.H.P.	INDICATED THRUST I.H.P. × 33000 ÷ (V. × 6080 / 60) ÷ A	EFFECTIVE THRUST E.H.P. × 33000 ÷ (V. × 6080 / 60) ÷ A	SLIP PERCENT P. × R. P.M. × 60 − V. × 6080 / P. × R. P.M. × 60	ADMIRALTY COEF. C_1 $V.^3 \times$ AREA ⊕ / I.H.P.	ADMIRALTY COEF. C_2 $V.^3 \times D.^{2/3}$ / I.H.P.
94	85	86	87	98	99	100	101	102	103	104
10								11.85		
11								13.03		
12	3280	1500	.4573	76.96	168.29	4.18	1.91	12.54	640.6	302.8
13	4430	1920	.4334	73.51	169.62	5.21	2.26	12.87	791.3	285.1
14	5700	2380	.4175	72.40	173.39	6.22	2.60	13.38	768.1	276.7
				72.65	173.62	7.27	3.03	13.85	754.2	271.7



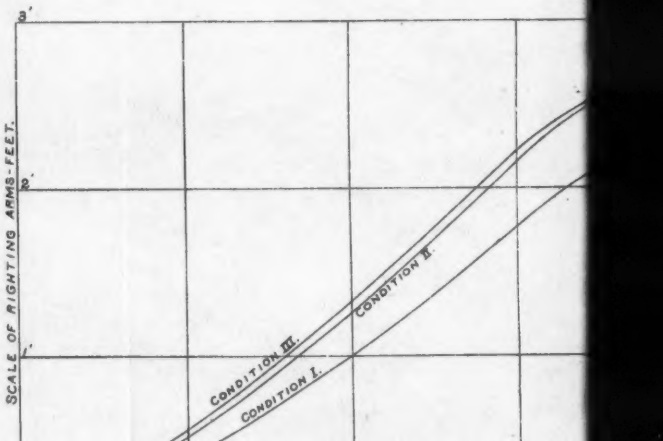
DATA TAKEN BY ENGINEERS DURING STANDARDIZATION

NUMBER AND DIRECTION OF RUN.	WHICH ENGINE.	STEAM PRESSURE. IN BOILERS ABOVE ATMOSPHERE.	STEAM PRESSURE. AT END ABOVE ATMOSPHERE	STEAM PRESSURE. M.P. STEAM CHEST. ABSOLUTE.	STEAM PRESSURE. IN 1st. RECEIVER ABSOLUTE.	STEAM PRESSURE. IN 2d. RECEIVER ABSOLUTE.	MEAN EFFEC. PRESSURE. H.P. CYLINDER.	MEAN EFFEC. PRESSURE. I.P. CYLINDER.	MEAN EFFEC. PRESSURE. FORWARD L.P. CYLINDER.	MEAN EFFEC. PRESSURE. AFTER L.P. CYLINDER.	MEAN EFFEC. PRESSURE. REFERRED TO LOW PRESSURE	VACUUM IN INCHES.	CUT OFF IN CYLS. FROM BEGINNING. H.P.	CUT OFF IN CYLS. FROM BEGINNING. I.P.	CUT OFF IN CYLS. FROM BEGINNING. FORWARD L.P.	CUT OFF IN CYLS. FROM BEGINNING. AFTER L.P.	TEMPERATURES. OUTSIDE AIR.	TEMPERATURES. ENGINE ROOM WORKING PLATFORM.	TEMPERATURES. F. ROOM WORKING LEVEL.	TEMPERATURES. INJECTION.	TEMPERATURES. DISCHARGE.	TEMPERATURES. HOTWELL.	TEMPERATURES. FEED.	REVOLUTIONS OR DOUBLE STROKES. CIRCULATING PUMP.	REVOLUTIONS OR DOUBLE STROKES. MAIN FEED PUMP.	REVOLUTIONS OR DOUBLE STROKES. AUX. FEED PUMP.	REVOLUTIONS OR DOUBLE STROKES. HOTWELL PUMP.	REVOLUTIONS OR DOUBLE STROKES. WATERSERVICE PUMP.	REVOLUTIONS OR DOUBLE STROKES. BILGE PUMP.	REVOLUTIONS OR DOUBLE STROKES. AUX. CONDENSER PUMP.	REVOLUTIONS OR DOUBLE STROKES. FIREROOM BLOWERS.	AIR PRESSURE IN INCHES OF WATER.	IN STARBOARD ENGINE. H.P. CYLINDERS.	IN STARBOARD ENGINE. I.P. CYLINDERS.	IN STARBOARD ENGINE. FORWARD L.P. CYLINDERS.	IN STARBOARD ENGINE. AFTER L.P. CYLINDERS.	IN STARBOARD ENGINE. TOTAL SUM COLS. 68, 69, 70 & 71.	IN P[illegible] H.P. CYLINDERS.
35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73
1-S.	ST'B'D. PORT																																342.55	222.92	149.56	136.45	851.48	209 ?
2-N.	ST'B'D. PORT																																543.9	438.37	253.27	271.13	1506.67	254 ?
3-S	ST'B'D. PORT																																736.1	670.6	334.5	357.7	2098.9	368. ?
4-N.	ST'B'D. PORT																																1659.8	1144.6	372.49	665.1	4041.89	1862.1
5-S.	ST'B'D. PORT																																1787.4	1740.7	673.65	893.06	5304.8	1807.6
6-N.	ST'B'D. PORT																																2314.7	2343.8	1250.2	1250.2	7158.9	2329
7-S.	ST'B'D. PORT	NO DATA.																															3433.5	2232.7	1680.6	1630.1	8036.9	2687
8-N.	ST'B'D. PORT																																2869.7	2999.7	1438.5	1481.4	8789.3	2618
9-S.	ST'B'D. PORT																																3241.0	3380.2	1640.1	1549.3	9810.6	2973
10-N.	ST'B'D. PORT																																3917.3	3683.6	1830.0	1874.1	11255	
11-S.	ST'B'D. PORT																																3884.5	3989.5	2455.7	2313.6	12753.3	CA
12-N.	ST'B'D. PORT																																4156.4	4358.6	2030.6	1972.6	12518.2	
13-S.	ST'B'D. PORT																																4070.8	4520.2	2104.2	2052.4	12747.7	

RESULTS OF CALCULATIONS FOR EFFICIENCY CURVES.

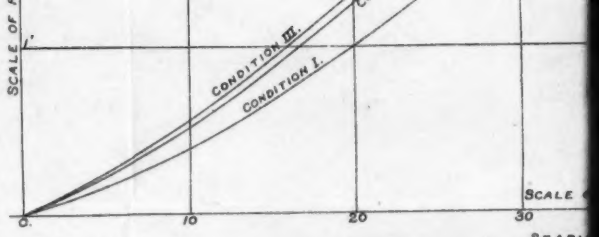
TRUE SPEED	TRUE I.H.P.	E.H.P.	PROPULSIVE COEF. E.H.P. / I.H.P.	REV. PER. MIN.^{3} / I.H.P.	REV. PER. MIN.^{3} / E.H.P.	INDICATED THRUST I.H.P. X 33000 ÷ V X 6080 / 60 + A	EFFECTIVE THRUST E.H.P. X 33000 ÷ V X 6080 / 60 + A	SLIP PERCENT P X R.P.M. X 60 - V X 6080 / P X R.P.M. X 60	ADMIRALTY COEF. C_1 V^3 X AREA ⊕ / I.H.P.	ADMIRALTY COEF. C_2 V^3 X $D^{\frac{2}{3}}$ / I.H.P.
84	85	86	87	88	89	100	101	102	103	104
10								11.85		
11								12.03		
12	3280	1500	.4573	76.96	168.29	4.16	1.91	12.54	840.6	302.8
13	4430	1920	.4334	73.51	169.62	5.21	2.26	12.97	791.3	285.1
14	5700	2380	.4175	72.40	173.39	6.22	2.60	13.38	768.1	276.7
15	7140	2970	.4160	72.25	173.69	7.27	3.03	13.85	754.2	271.7
16	8720	3730	.4278	72.69	169.93	8.33	3.56	14.21	749.4	270.0
17	10440	4640	.4445	73.74	165.91	9.38	4.17	14.56	750.8	270.5
18	12360	5720	.4628	74.87	161.76	10.49	4.84	14.91	755.8	272.2

CU



14	5700	2380	.4175	72.40	173.39	6.22	2.60	13.38	768.1	276.7
15	7140	2970	.4160	72.25	173.69	7.27	3.03	13.85	754.2	271.7
16	8720	3730	.4278	72.69	169.93	8.33	3.56	14.31	749.4	270.0
17	10440	4640	.4445	73.74	165.91	9.38	4.17	14.56	750.8	270.5
18	12360	5700	.4612	74.87	162.36	10.43	4.84	14.94	752.8	271.2
19	14540	6960	.4787	76.25	159.30	11.69	5.60	15.45	752.7	271.2
20	17070	8520	.4992	77.76	155.79	13.04	6.51	16.18	747.7	269.4
21	20060	10490	.5224	79.76	152.68	14.61	7.63	17.33	735.9	265.1
22	23200	13060	.4985	75.26	150.99	18.20	9.07	19.19	648.4	233.6

NOTE. V = SPEED IN KNOTS., A = PROJ. AREA OF SCREWS., P = PITCH OF SCREWS., D = DISP. IN TONS., ℳ = MIDSHIP SECTION.



CONDITIONS.
I. SHIP LIGHT - NO CREW, COAL, AMMN. STORES OR FRESH WATER ON BOARD.
II. SHIP IN NORMAL CONDITION. TRIAL ALLOWANCE OF COAL, STORES &C, ON BOARD.
III. SHIP FULLY LOADED.

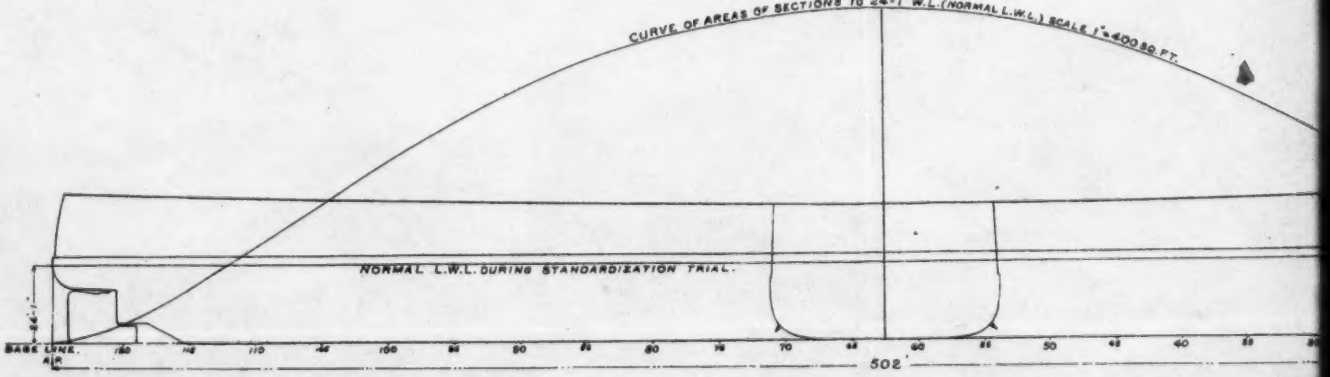
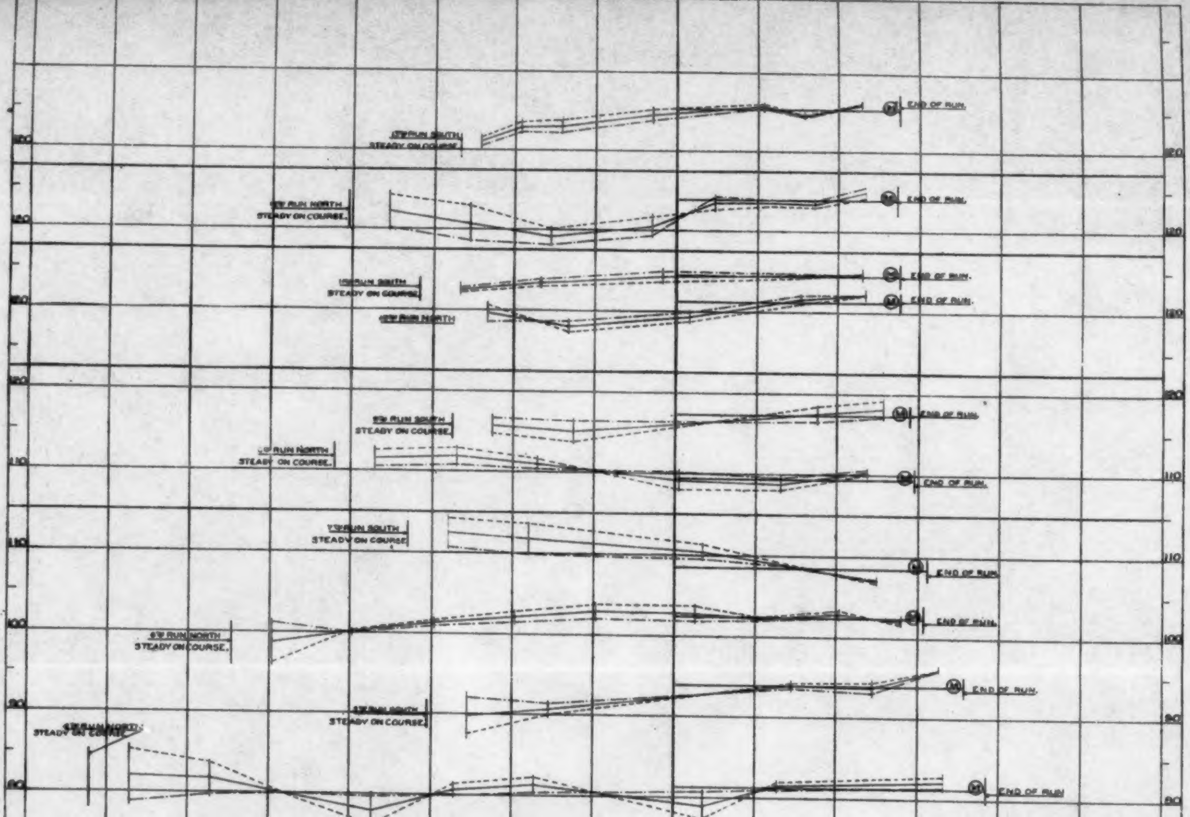
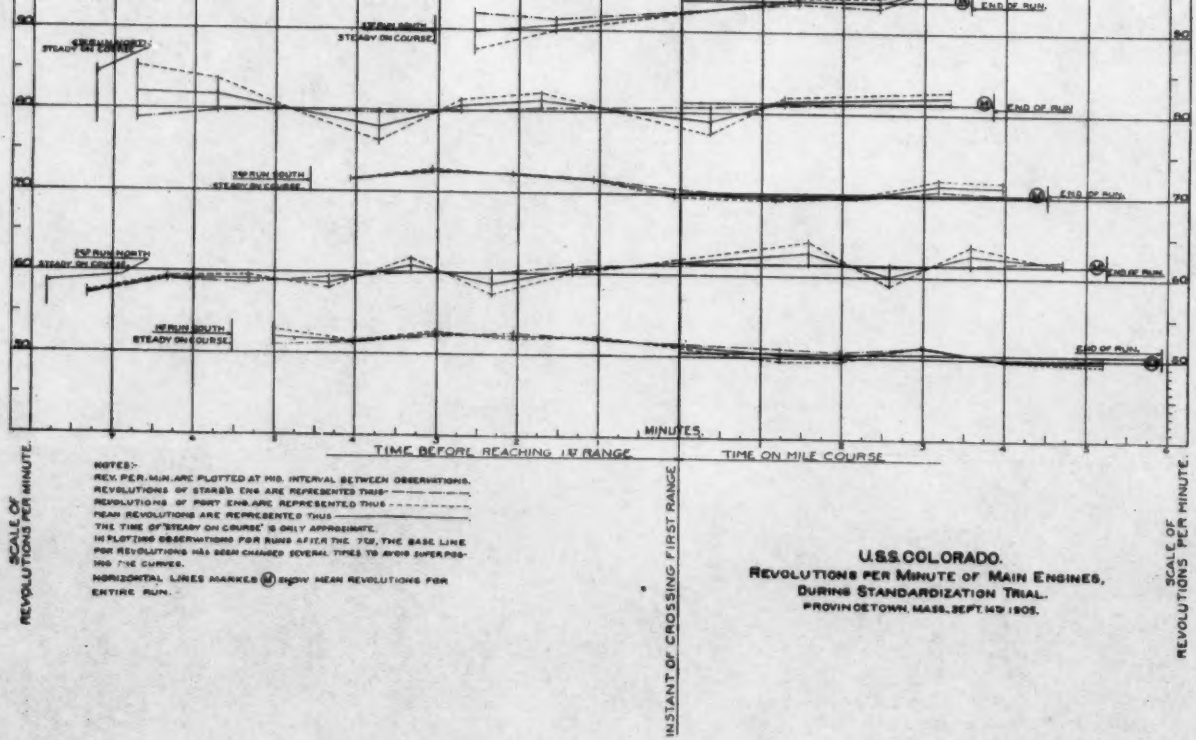


PLATE [illegible]





NOTES:-
REV. PER. MIN. ARE PLOTTED AT MID. INTERVAL BETWEEN OBSERVATIONS.
REVOLUTIONS OF STARB'D. ENG ARE REPRESENTED THUS ——— ——— ———.
REVOLUTIONS OF PORT ENG. ARE REPRESENTED THUS - - - - - - - - - -.
MEAN REVOLUTIONS ARE REPRESENTED THUS ———————.
THE TIME OF "STEADY ON COURSE" IS ONLY APPROXIMATE.
IN PLOTTING OBSERVATIONS FOR RUNS AFTER THE 7TH, THE BASE LINE FOR REVOLUTIONS HAS BEEN CHANGED SEVERAL TIMES TO AVOID SUPERPOSING THE CURVES.
HORIZONTAL LINES MARKED (M) SHOW MEAN REVOLUTIONS FOR ENTIRE RUN.

U.S.S. COLORADO.
REVOLUTIONS PER MINUTE OF MAIN ENGINES, DURING STANDARDIZATION TRIAL.
PROVINCETOWN, MASS., SEPT. 14TH 1908.

Fig. 1.
BACK BEACON EAST RANGE,
"PROVINCETOWN."

Fig. 3.
VIEW EAST RANGE FROM COURSE,
"PROVINCETOWN."

Fig. 4.
BACK BEACON NORTH RANGE,
"ROCKLAND."

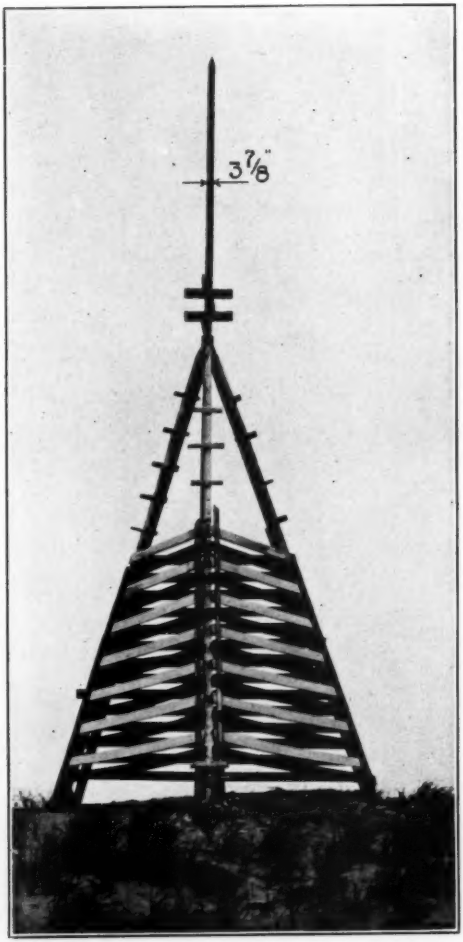


Fig. 2.

FRONT BEACON EAST RANGE, "PROVINCETOWN."

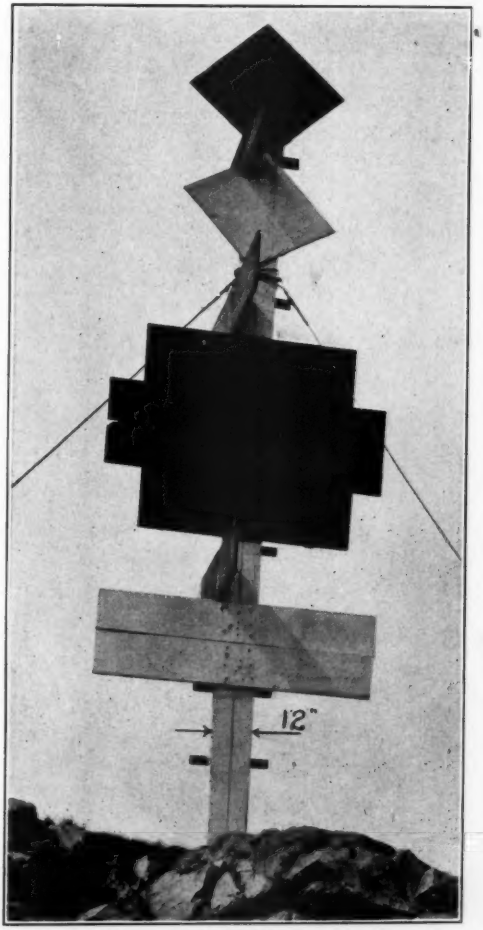
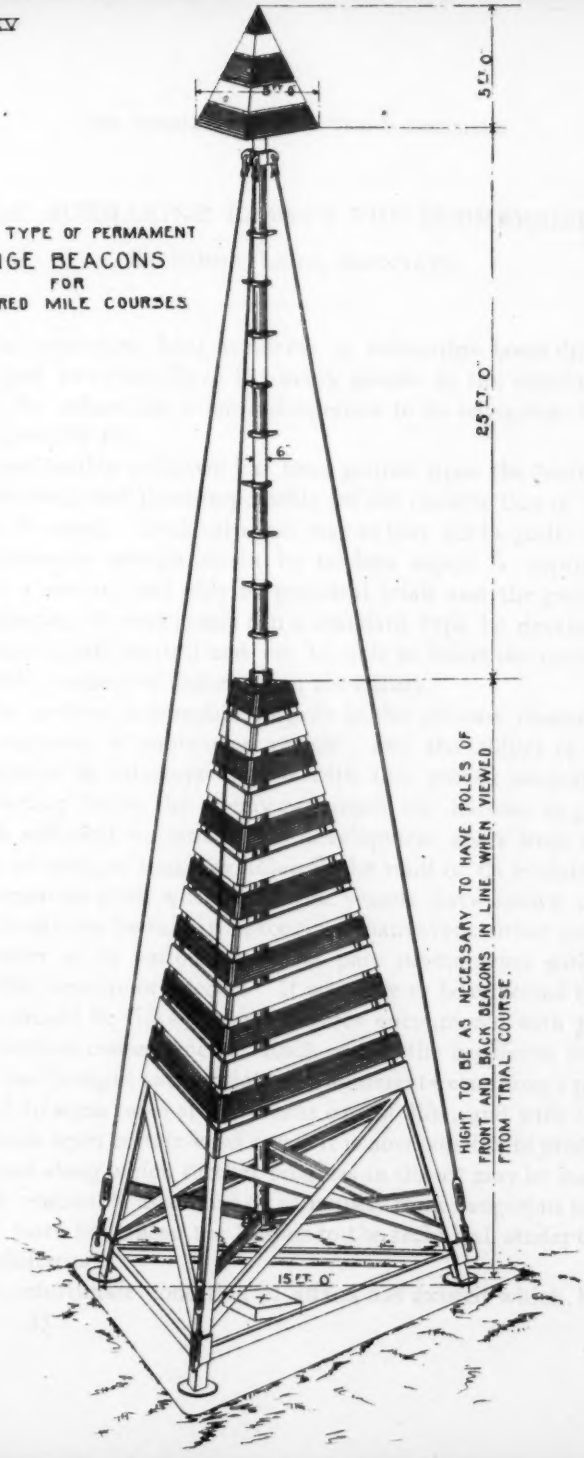


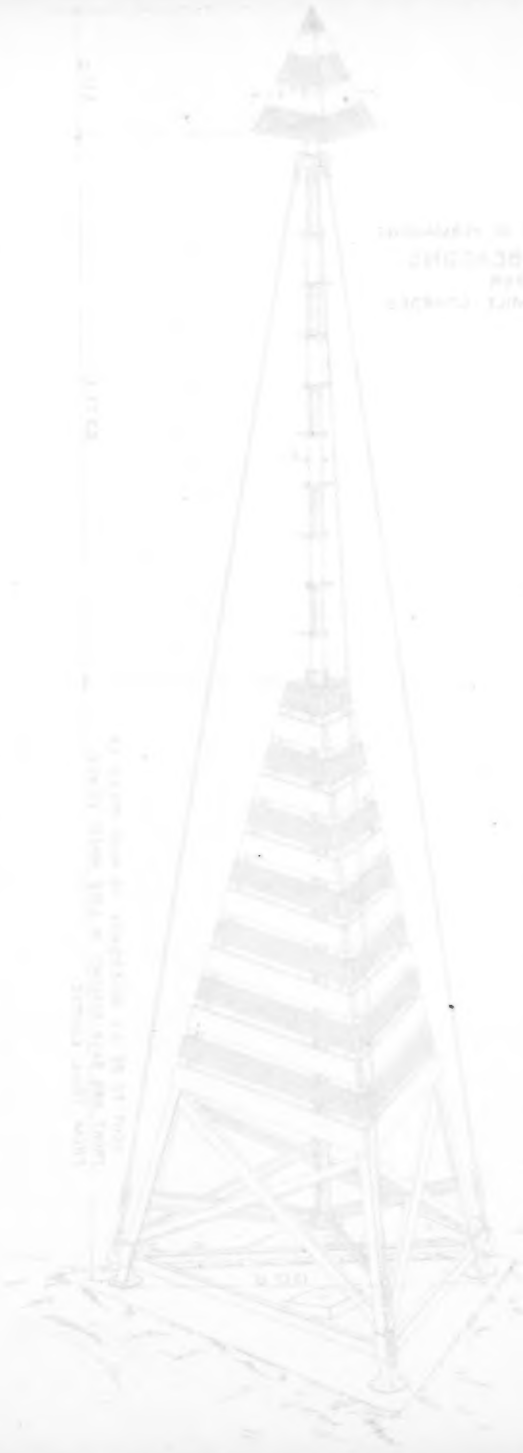
Fig. 5.

FRONT BEACON NORTH RANGE, "ROCKLAND."

PLATE XV

PROPOSED TYPE OF PERMAMENT
RANGE BEACONS
FOR
MEASURED MILE COURSES





THE SUBMARINE *VERSUS* THE SUBMERSIBLE.

By Simon Lake, Associate.

The numerous fatal accidents to submarine boats during the past two years have led many people to the conclusion that the submarine is more dangerous to its navigators than to a possible foe.

Considerable criticism has been poured upon the heads of departments and those responsible for the construction of this class of vessel. Such criticism may or may not be justly due.

Submarine navigation in its modern aspect is comparatively a new art, and only by practical trials and the gradual elimination of weaknesses can a standard type be developed capable of safe control and yet be able to inflict the greatest possible measure of injury upon the enemy.

The modern automobile torpedo is the primary reason for the existence of submarine vessels; and the ability of the submarine to intercept the foe with this proved weapon of destruction before the enemy can reach our harbors by gunfire is sufficient warrant for its development apart from any other advantages that may follow in the train of its evolution.

Numerous trials with submarine vessels have shown that such boats can be run, in peace-time maneuvers, either under the water or in various stages of part submergence with a variable measure of success. It was only to be expected that there should be risks run and failures encountered with possibly serious consequences. Each one of the accidents, however, has brought substantial enlightenment even though purchased in some cases at a grievous cost of life; and with this emphasis upon certain weaknesses it is now possible to predict the lines along which further progress in the art may be made with a reasonable assurance of immunity from dangerous mishap. Such have been the lessons to the technical student of the submarine.

An unfortunate condition of affairs has existed which has

prevented a full understanding of the subject on the part of those responsible for the purchase of many of the submarine boats now extant: this has been the veil of secrecy cast about the submarine question by most of the inventors and builders of under-water craft. The inventors and builders have been compelled to keep secret many things which they have learned —perhaps by costly experimentation—at least until such time as patent protection could be secured. Many other features, the fruit of experience, but on which no such protection could be secured, have been kept secret for the soundest of business reasons. The writer has been no exception in these particulars. The fatal accidents of the past two years, however, have prompted him to make public the results of certain experiences and investigations, with a view to contributing, as far as lay in his power, to the prevention of disasters kindred to those that have so shocked the civilized world and which in the eyes of the general public have cast discredit upon the art of submarine navigation.

Submarine vessels are broadly distinguished today by the two denominations of "submarine" and "submersible." The difference between these terms has not always been well defined. Boats of each type naturally are submarines in the common understanding of that word. The term "submersible" was first given to the *Narval*, built in France in 1899. The earlier types of French boats were of the diving or plunging class originally; while the *Narval* was the first of the French boats primarily designed to *submerge* or *be submerged* bodily by the down-pull of hydroplanes rather than to *plunge* or *dive* by the head, as is common in the diving type. It is in these distinctions that lie the most important differences among the present types of under-water boats.

France, to whom we are thus indebted for the term "submersible," is the only country in which the study and the development of submarine craft has been entirely under governmental auspices. There has been an entire absence of a spirit of commercialism in that investigation; and the conclusions of the French Admiralty today in favor of the "sub-

mersible" are based upon serious, painstaking and practically exhaustive researches which have candidly considered submarine craft of all forms and types.

Russia, under the stress of war, being uncertain as to which was the type most to be desired, purchased boats of several sorts in order to determine practically the class best fitted to meet the demands of actual conflict. None of the boats ordered were delivered in time to be factors during the period of hostilities; but the circumstances of war taught its lessons and showed the Russians the conditions that must be met by a successful submarine craft. Maneuvers and operations since undertaken in the light of these lessons by her active submarine flotillas have served to verify the conclusions of the French and to add Russia's decision in favor of the "submersible" as that term is understood in this article.

England and the United States have tried only one type—that of the diving order; and it is the diving submarine that has met with so many of the accidents—some of them fatal—during the past two years. Let us see if the reasons for these mishaps cannot be made clear.

The *sine qua non* to the successful working of the diving boat is extreme sensitiveness in a fore-and-aft direction when submerged, or in other words, *lack of longitudinal stability.* Without this tenderness it is practically impossible, at the speeds so far attained, to control such a vessel when running submerged.

It is not necessary here to discuss either the administrative or the military aspects of this lack of longitudinal stability. The accidents that have occurred make a more direct and human appeal by emphasizing the dangers faced by their crews even in times of peace.

The submerging or "submersible" boat requires no such hazardous limitation in the direction of safety in order to make her operatively a success. In the "submersible" it is possible to secure a far greater measure of longitudinal stability than is possible in the diving class. This greater longitudinal stability which may be secured in the "submersible"

does not in any sense affect the speed and precision of action either in submerging or in maintaining a uniform degree of submergence when once under the surface. All that this may mean from a military or an administrative point of view has been thoroughly discussed elsewhere. Let us see, however, what are the physical conditions operating when a boat is submerged and against which those in control of a "diving submarine" must effect a balance by skill either in the form of deftness of handling or nicety of adjustment in one direction or another.

The loss of the British submarine *A8*, therefore, may be taken as a case exemplifying in the plainest and most startling manner the dangers peculiar to the diving submarine. The testimony taken during the court-martial showed that that boat when running in light condition had a normal reserve of bouyancy of about 16 tons. This had been reduced to 6 tons, and she was so trimmed that she was *down four degrees by the stern* at the time of the accident. The boat was running at a speed of ten knots under her gasoline engines. It was further shown, by the testimony of Captain R. H. Bacon, R. N., that a moment to change trim of no less than 132 foot-tons would be required to bring the boat horizontal or on an even keel when trimmed in the condition described, and that *a still greater moment to change trim* would be required to bring the vessel down by the head so as to make her plunge. The practical proof that sufficient force *was* exerted in some manner to effect this very end was given by the sudden and disastrous manner in which that boat dived.

Both Sir William White and Captain Lees, R. N., have recorded their belief that the *form of hull* of the *A8*, combined with her limited longitudinal stability, was in itself the prime reason for the sudden plunging of that vessel. Model experimental trials, both in the United States and in Germany, have only served to confirm that assumption. As far back as 1897, towing experiments with my boat *Argonaut I*, in her original form, made plain this tendency of the cigar-shaped hull, even when united with great fore-and-aft stability, to

bury or to run under by the bows. The *Argonaut I* was subsequently modified by the addition of a ship-shaped superstructure which eliminated completely the down-pull at the bows previously so markedly present.

Without referring to the detailed results of my model-tank experiments, the elemental difficulties can be shown in a graphic manner with sufficient accuracy to demonstrate the forces at work tending to unbalance or upset a craft of the diving type *already* more or less unstable in a fore-and-aft direction and in which the reserve of buoyancy is needfully reduced to a dangerous degree that the boats may be made operative at moderate speeds.

The following diagrams will help to make the circumstances clear to the general reader. In Fig. 1, we have a boat of the diving type running submerged in the usual manner, that is, with a trim of from two to four degrees by the head in order to enable the vessel to remain under water. In the diagram, "A" represents the position of the center of buoyancy, "B" the center of gravity of the vessel, and "C" the horizontal rudder which is to overcome the righting moment of the short arm between "A"and "B".

Experience and model experiments have shown that the down-pull at the bow in a cigar-shaped craft is much greater at or near the surface and at or close to the bottom; and it has been found that this down-pull changes with the depth below the surface or the distance above the water bed. To run "awash", therefore, it is necessary to trim by the stern, and still further to offset any tendency to plunge by setting the horizontal rudder a number of degrees "*to rise*", as was the case with the *A8*.

To dive, additional water is admitted into what is called the forward trimming tank, the horizontal rudder "C" is set to "dive", and the vessel plunges under, head first, like a porpoise. Now, the trick in handling a diving boat is to allow her to reach a depth of several feet below the surface—where the down-pull at the bow is less—and then to catch her and hold her at that submergence by the *continual* manipulation of the horizontal rudder, "C".

The difficulties of the problem may be better understood by referring to Fig. 2, in which the conditions prevailing are diagrammatically likened to a balanced beam, for which the pivitol point "A" corresponds to the center of bouyancy of the vessel, and the weight centered at "B," acting as a short pendulum, corresponds to the weight of the vessel as a whole. "B", then, is the one force tending to hold the beam horizontal. "H" is a float, representing the amount of reserve buoyancy statically in the craft when trimmed to run submerged, and bears to the total displacement of the boat the ratio of *only about 1 to 560* in the case of vessels of the "*A* class" of 200 tons. It must be plain that this very small reserve of buoyancy can add practically nothing to the stability of the beam, which we assume is floating. "F" is a weight applied at the short arm of the beam and represents the downward pull at the bow *graphically but not relatively*, because this weight or force is a *continually changing* quantity, which must be as continually checked or counterbalanced by the force of the stream lines impinging against the horizontal rudder "C"—in the direction of "T" which must equal, in effect, a counterbalancing weight placed at "C" and acting in the direction of "T". With expert steersmen, and by fine adjustments of water ballast, a certain measure of success has been won after experimenting with vessels of this particular type for a period of half a century. But this success is at the expense of *constant vigilance*, and with so many restrictions that under some conditions it is still impossible or dangerous to navigate submerged in this manner.

Again referring to Fig. 1, it will be seen that when the boat once reaches the position shown she is running with a slight downward inclination by the bow, which is required to balance the upward pull of the reserve buoyancy of the vessel. In smooth water and with the vessel running at an angle of only a few degrees, the stream lines "D" and "E" are parted at the bow easily and sweep around the vessel in an approximately uniform manner. In this condition, the boat moves forward in a plane parallel to the surface of the water, but *not*

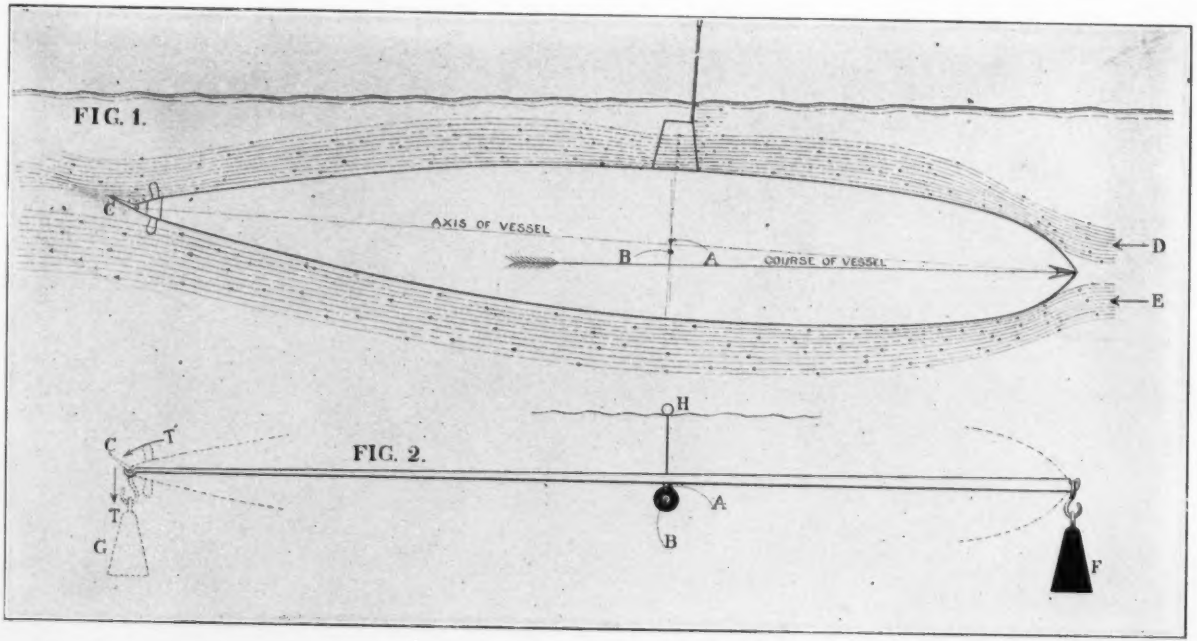


FIG. 2.

with the axis of the craft parallel to the surface. In other words, the *boat does not follow the true direction of her axis, but* moves forward at an angle, and her course is a mean between two opposing forces: first, that of buoyancy, which is continually tending to bring the vessel to the surface, and secondly, that force due to the downward thrust of the screws in line of the axis of the boat. These opposing forces can be balanced only by exceptional skill on the part of the man at the diving rudder; otherwise the maintenance of a uniform depth in a diving boat becomes impossible.

In addition to the two forces just mentioned, there are others which tend to destroy the trim of the boat. The first of these is the righting moment of the pendulum, "A–B", which tends to bring the vessel on an even keel. This may be considered as a constant force, but for the shortness of its arm it can have, of necessity, but a relatively slight effect. It is, however, the one constant force, and the *only one* which prevents the craft from swinging like a pin-wheel around its fulcrum at the center of bouyancy, "A". The next disturbing influence is that of the down-pull at the bow, already noted, which varies according to the speed of the boat and the depth of submergence. The fifth element of disturbance is that of the subsurface effects of wave motion, which gives a varying head, and, consequently, intermittent change of pressure or down thrust on different portions of the boat. The sixth force is one within the craft herself, and is the result of shifting weights in the form of surging water ballast, changes in position of the crew, or the transfer fore or aft of other masses.

It will thus be seen that there are six disturbing forces to be taken into account, and of these four of them are continually fluctuating. These opposing forces must be harmonized by the manipulation of the horizontal rudder, "C", which, by the exercise of great skill and control and within prescribed limits, it has been found possible to do under certain chosen conditions. A moment's inattention, however, on the part of the helmsman, or the failure of the horizontal rudder to

respond with sufficient quickness, may lead at once to loss of control; and the vessel will either break for the surface and broach, or plunge toward the bottom, with the risk of disaster under the impulse of her driving engines and the rapidly increasing inclination of her bow. If the engines can be stopped before the vessel has struck the bottom, then the two constant forces—those of the righting moment of the pendulum "A–B" and that of the reserve bouyancy—will combine to bring the vessel back to her normal horizontal position and to draw her again to the surface. It should be plain from this what must be the task of the operator at the diving rudder, and to what degree his skill must be developed to meet the conditions thus confronting him even in time of peace and amid surroundings and circumstances carefully chosen. How much greater, then, must be his task when operating amid the environments of warfare, and under the nervous stress of actual conflict. Sub-Lieutenant Godfrey Herbert, one of the officers aboard the *A4* when she sank, testified, as stated in the "Naval and Military Record :"

"The boat sank to a depth of ninety feet before starting to rise. During this time water was coming down the ventilator with considerable force, which increased as the depth got greater. In about a minute the lights went out, and it was with great difficulty that the Captain's order to put the 25-H.P. pump onto the fore bilge was attempted. Just before the lights went out the boat had an *inclination by the bow of forty degrees*, and all small gear and the cloth covering the batteries were washed forward. * * * * The gauge was shown at ninety feet. *It could not show more.*"

From an editorial in the same paper we find the following:

"They are liable," says Sir William White, "to unexpected dives, and this *again is confirmed in the case of the A4 by the testimony of the cockswain.*"

Fig. 3 represents a boat of the diving type and of the *A* class submerged and inclined at *just half the angle* assumed by the *A4* when she sank. It will be seen from this diagram how considerably the down-pull at the bow is augmented by

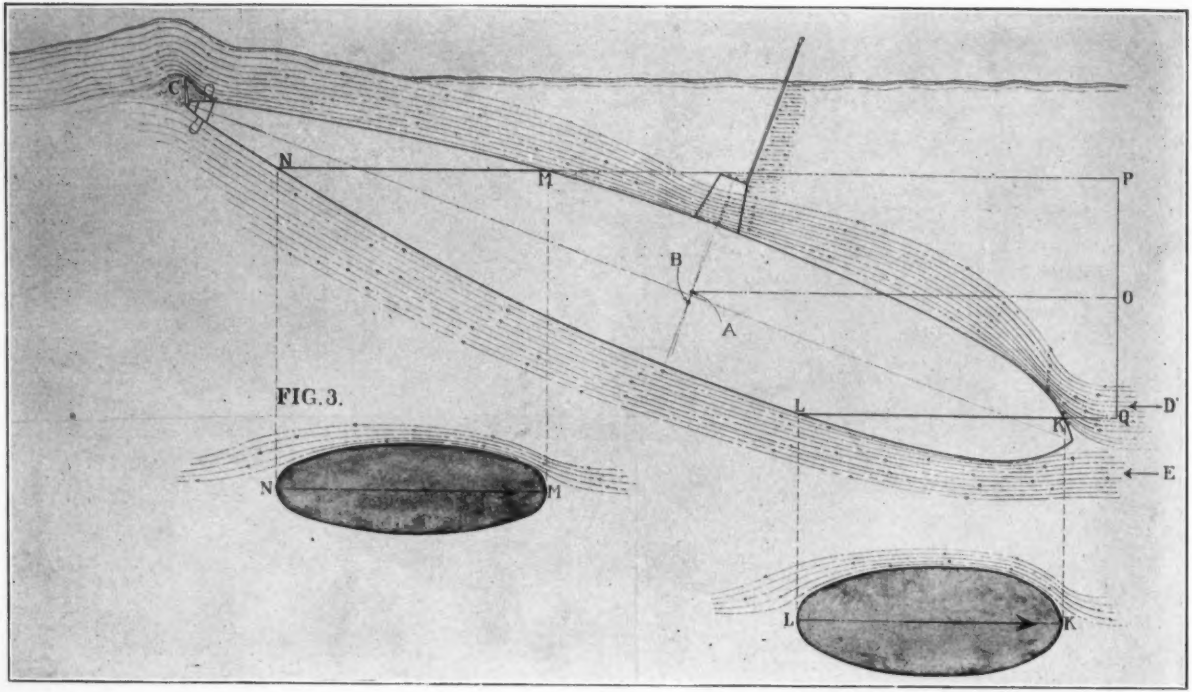


FIG. 3.

every change of trim on the part of the cigar-shaped craft; and it is not difficult to understand how a very rapid increase of inclination may render less effective the counterbalancing factor of the horizontal rudder, "C". This has been borne out by numerous official reports.

In the case of Fig. 3, it is assumed that the boat has started in the condition primarily required—with trim by the head—as indicated in Fig. 1, and that the vessel was then moving forward in a plane parallel to the surface. By reason of some of the causes or forces heretofore described, the craft now takes a further inclination or dip by the bow. Fig. 3 shows graphically what occurs when the momentum of the vessel tends to maintain her on her forward course in a plane parallel to the surface. The stream lines, instead of parting fairly on each side of the bow, come with greater force against the forward upper half of the vessel and impart, by reason of this, a strong downward thrust at that point, while the supporting resistance on the under side of the hull at the bow is considerably lessened. Consequently, the balance is destroyed and a very much larger angle of the horizontal rudder, "C", is called for. There is, of course, a limit to the effectiveness of the horizontal rudder, "C". A horizontal section drawn through "L–K" and another drawn through "M–N", each equally distant above and below the center of bouyancy when the boat is inclined as shown in Fig. 3, give the relative *areas of resistance* and illustrate how much more rapidly the forward or lower area increases than the after or upper area. It will be seen from these sections how much more difficult the lower one is to move forward through the water than the after or upper section. This, of course, is due to the very bluff character of the lower section and its greater tendency to produce eddying, back pressures, etc., when driven at a considerable speed. Therefore, the condition becomes as follows: The vessel moves forward, first in a horizontal plane, and then, for some of the reasons previously stated, additional inclination occurs; resistance to progress of that part of the vessel below the swinging point, "O", (*i. e.*, a line intersecting the center of bouyancy) increases and

retards the bow while the momentum of the craft carries the upper and less resisting section, now the tail end or afterbody of the boat, forward; any shifting weights, providing the angle has become great enough, would move forward, as in the case of the *A4;* the horizontal rudder, even when given its maximum angle of rise, proves ineffective; and the boat continues to dive with a rapidly increasing inclination unless the machinery is stopped and the vessel's momentum arrested. If the vessel during this interval has not been brought to a standstill by the bottom she may continue to plunge to a depth where the pressure may prove sufficient to cause her collapse.

Fig. 4 illustrates a "*submersible*" of the latest type, which will be explained in connection with diagram 5, so that a direct comparison may be drawn with the working principles of the *diving* boat as shown in Figs. 1, 2 and 3.

In Fig. 4, the "*submersible*" is shown running on a level keel—the position normal to boats of this type—instead of being several degrees down by the bow, as is necessary with the "*diving*" type. The reason why it is thus possible to run the "*submersible*" on an even keel is as follows:

It will be observed, in the first place, that the hull is quite different in form. This raises the center of buoyancy to a point much higher than it is possible to do in the regulation cigar-shaped craft of the "*diving*" type. The vessel is also provided with a heavy keel which lowers the center of gravity. Therefore, the arm of the pendulum, "A–B", becomes longer and its righting moment correspondingly greater. This is a most important feature, as it contributes directly to securing the *greatest possible measure of longitudinal stability* under the circumstances. As the boat does not need to be inclined to submerge her, there is no limit—*by reason of operative requirements*—to the longitudinal stability that may be given the "*submersible*"; and in practice it is desirable, of course, to have the longitudinal stability as great as the form of hull and the arrangement of weights will permit.

To run "awash" or submerged, the vessel is trimmed by

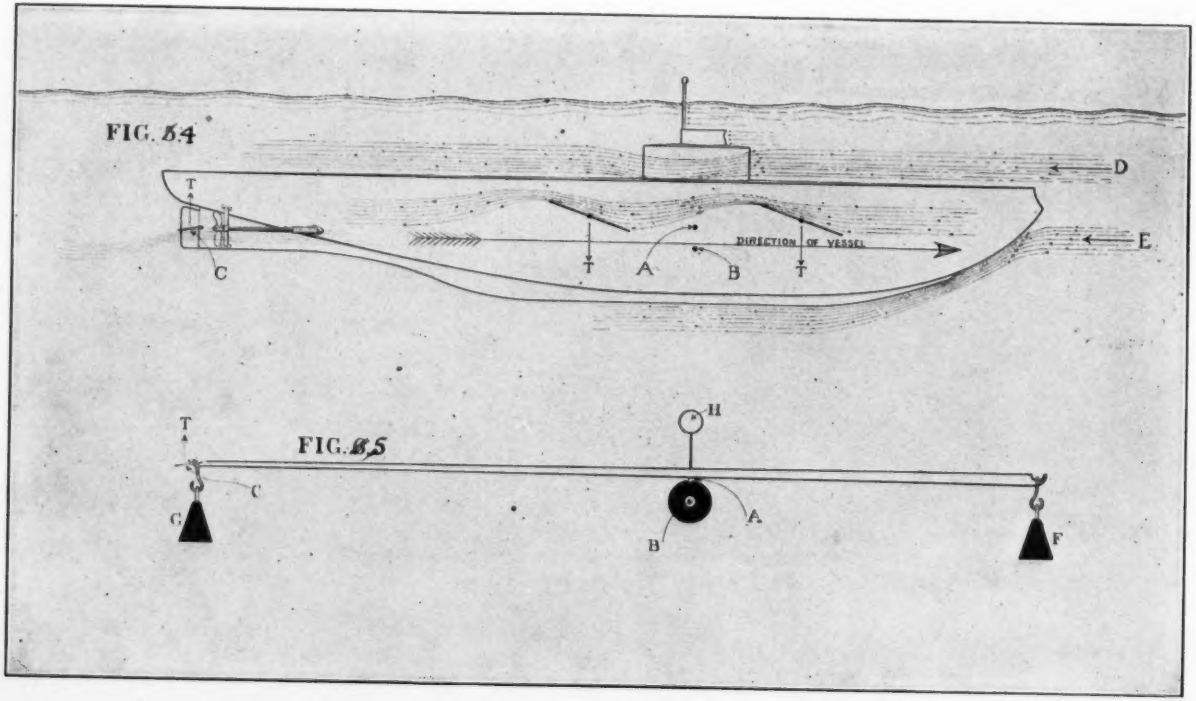


FIG. 54
T
C
T
A
B
T
DIRECTION OF VESSEL
D
E
T
FIG. 55
H
C
G
B
A
F

adding water ballast until only the desired amount of reserve buoyancy remains which will ensure the boat resting on an even keel with her conning tower in an awash condition in relation to the normal surface. In other words, the beam or balance represented by Fig. 5 is brought to a horizontal position by adding *fixed weights* at *both ends* so that "O" and "F" are equal. In this condition the "*submersible*" can be run without risk of or impulse toward a sudden dive.

The official reports and the testimony given during the court-martial following upon the loss of the *A8* confirm the belief that it is either impossible or dangerous to run "*diving*" boats upon the surface on a level keel even when retaining a very considerable reserve of buoyancy. This hazard is primarily due to the down-pull at the bow characteristic of the cigar-shaped form of hull and lacking longitudinal stability of vessels of the "*diving*" type. With the "*submersible*" of the type illustrated here there are none of the tendencies to dive so characteristic of the other order of boat under these circumstances. The reasons for this are the great longitudinal stability and the structural form which completely offsets the development of a downward pull.

The vessel being in the awash condition just described, is given headway, and it is soon noticed that there is a slight tendency to change trim by *rising at the bow*, due to the thrust against the large conning tower, and this is at once corrected by a *permanent* angle of the horizontal rudder, "C", intended for this particular purpose. It will be observed, of course, that this horizontal rudder is placed aft as in the "*diving*" boats, but it is used for the very *opposite* purpose of keeping the vessel in a horizontal position rather than to cause her to take a dip or plunge, as is the case with the "*diving*" type. A downset of the horizontal rudder of only two or three degrees is usually sufficient to balance the tendency of the bow to *rise* in boats of the "*submersible*" type herein illustrated. A level keel being thus secured, the angle of the horizontal rudder remains constant for a given speed.

The "*submersible*", still running in the awash condition

but ready for instant submergence, is quickly brought completely under the water, without change of trim, by depressing the forward ends of the hydroplanes. This causes the stream lines at the side of the boat to impinge against the upper surfaces of the hydroplane and to give a resultant downward thrust as indicated by "T", "T", in Fig. 4. The hydroplanes being four or more in number and symmetrically disposed in regard to the center of buoyancy or fulcrum, the thrust against their upper faces causes the boat to sink on an even keel. It will be observed that the stream lines at the bow are not affected in the same manner as those at the bow of the "*diving*" boat of cigar-shaped form. There are none of the disturbing eddies developed peculiar to the spindle-shaped hull; and the vessel moves forward along the line of least resistance instead of at an angle—the thrust of the screws or the momentum of vessel when the machinery is stopped impel her in the direction of her major axis, *which is in the plane of the course desired.*

When the vessel has reached the depth desired, the hydroplanes are brought back to an angle which will yield just pressure enough to counterbalance the upward pull of the reserve bouyancy statically in the craft and indicated by "H". Because of the *buoyant* character of the superstructure and the greatly increased buoyant moment of the conning tower in vessels of this class the righting effect of this force is very much greater than can possibly be the case with boats of the "*diving*" type. The reserve buoyancy is the *only force* which it is necessary to counteract in "*submersibles*" of the type under discussion; and this force is a *constant* one and so easily balanced or neutralized that the writer has frequently left the controlling mechanism of the hydroplanes—located in the conning tower—and has gone below into the engine room for periods of several minutes at a time, and has returned to the conning tower and found the boat running still at substantially the same depth as that at the instant of his departure. The reason for this is the *fact* that in the "*submersible*" we thus do away with *four* fluctuating forces

which are inherent or peculiar to boats of the "*diving*" type and spindle form of hull.

The practical results thus secured by the "*submersible*" are:

1st. Safety in submerged control.

2d. Facility in handling that calls for no special training or rare dexterity on the part of a single member of the crew.

3d. Easy maintenance of a uniform depth on an even keel in waters of variable depths, thus making it safe and practicable to operate either in deep or shallow waters.

4th. The removal of restraint to free movement on the part of the crew in the *very fullest* discharge of their duties.

5th. By preserving an even keel, insuring the effective operating of the observing instrument the moment it appears above the surface, and thus affording that measure of safety and efficiency following in the train of rapid observation of the entire horizon.

6th. The ability to discharge torpedoes with the vessel always on an even keel, *thus insuring the maximum probability of making a successful target.*

These advantages have appealed with convincing force to both the French and the Russians, who have examined and tested exhaustively these rival methods of operating; and the practical reasons for this decision are not to be denied, while the records of so many serious accidents to "*diving*" boats in the hands of *skilled crews* give us cause for pause and question.

THE LOVEKIN IMPROVED INBOARD COUPLING FOR LINE AND PROPELLER SHAFTS.

By Luther D. Lovekin, Associate.

In view of the growing demand for a reduction in weight and cost in all propelling machinery, and particularly in naval vessels, the author has given his best attention to the designing of a series of loose couplings for propeller and line shafting, which combine extreme simplicity with maximum strength.

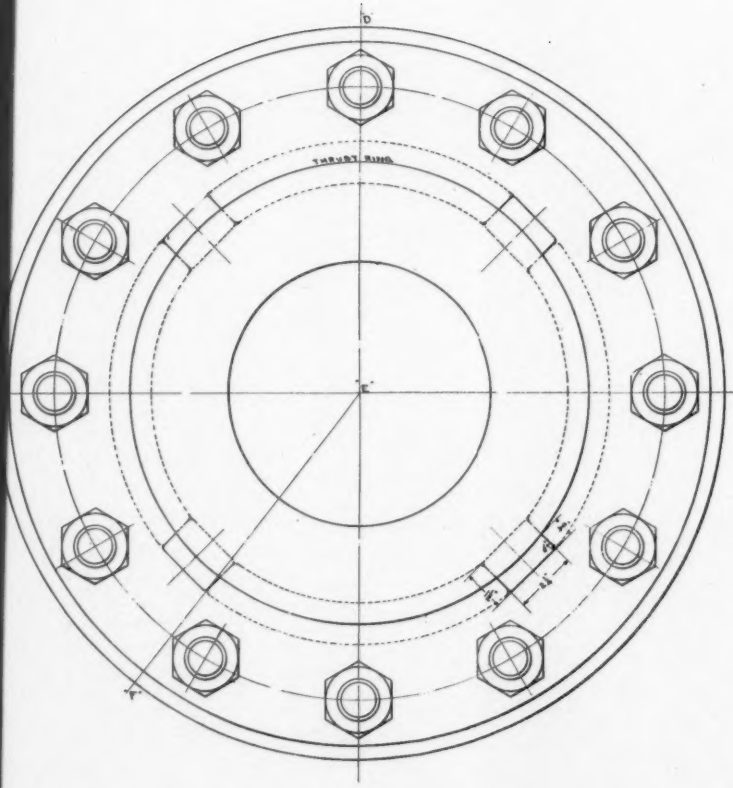
One of these couplings was fully illustrated in the Journal of the American Society of Naval Engineers, Volume XVII, No. 2, by Robert S. Riley, with a full set of calculations of the same, and also a comparison of weights with former couplings.

It is my purpose in this article to consider a type being made for the U. S. battleship *New Hampshire*, in order to show the great economy in weight and cost to be obtained by the use of the *New Hampshire* type over that of the *Kansas* type, particularly where a high-grade material is used, such as in naval work, where the cost per pound runs into very large figures.

It will be noticed that this coupling possesses many advantages, and in order to bring them out in point of merit, I append the following list of advantages, together with a full set of calculations for arriving at a design wherein the coupling is of the same strength as the adjoining shaft, which is taken as the unit for design.

LIST OF ADVANTAGES OF THE LOVEKIN IMPROVED INBOARD LOOSE COUPLING (PATENTED) FOR U. S. BATTLESHIP "NEW HAMPSHIRE."

1. The keys for transmitting torsion are fitted the entire length of the loose collars, thus utilizing the space formerly wasted.



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N THE GROOVE OR RECESS AND THE END OF
D PROPORTIONAL TO THE STRENGTH OF
D RECESS.

2- THAT PORTION OF KEY BEYOND THE RING NEED ONLY BE OF SUFFICIENT LENGTH TO TRANSMIT THE TORSION REQUIRED TO MAKE UP THE FULL TORSIONAL STRENGTH OF THE ADJOINING SHAFT, THIS IN MOST CASES IS THE LINE SHAFT, AND AS SUCH, IT IS ALWAYS MADE OF A LESS DIAMETER THAN THE PROPELLER SHAFT, WHICH ENABLES US TO DESIGN THE COUPLING IN THE MANNER DESCRIBED WITHOUT ADDING ANY ADDITIONAL METAL TO THE INSIDE OF THE SHAFT WHICH IS GROOVED.

3- THE COUPLING BOLTS ARE DESIGNED SO AS TO TRANSMIT THE ENTIRE SHEARING LOAD AT THE JOINT 'A' AND TO TRANSMIT SUCH PROPORTIONS OF THE TOTAL LOAD AT THE JOINT 'B' AS MAY BE REQUIRED BY THAT PORTION OF THE KEY BEYOND THE RING, THIS MAKES IT ABSOLUTELY NECESSARY TO USE A TAPER BOLT AND THE SMALL END OF SUCH BOLT NEED ONLY BE PROVIDED WITH THREADS OF SUFFICIENT STRENGTH TO WITHSTAND THE THRUST OF THE PROPELLER WHEN GOING ASTERN WHICH IS VERY SMALL WHEN COMPARED WITH THE ACTUAL STRENGTH OF THE SHAFT PROPER.

4-
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KEYS
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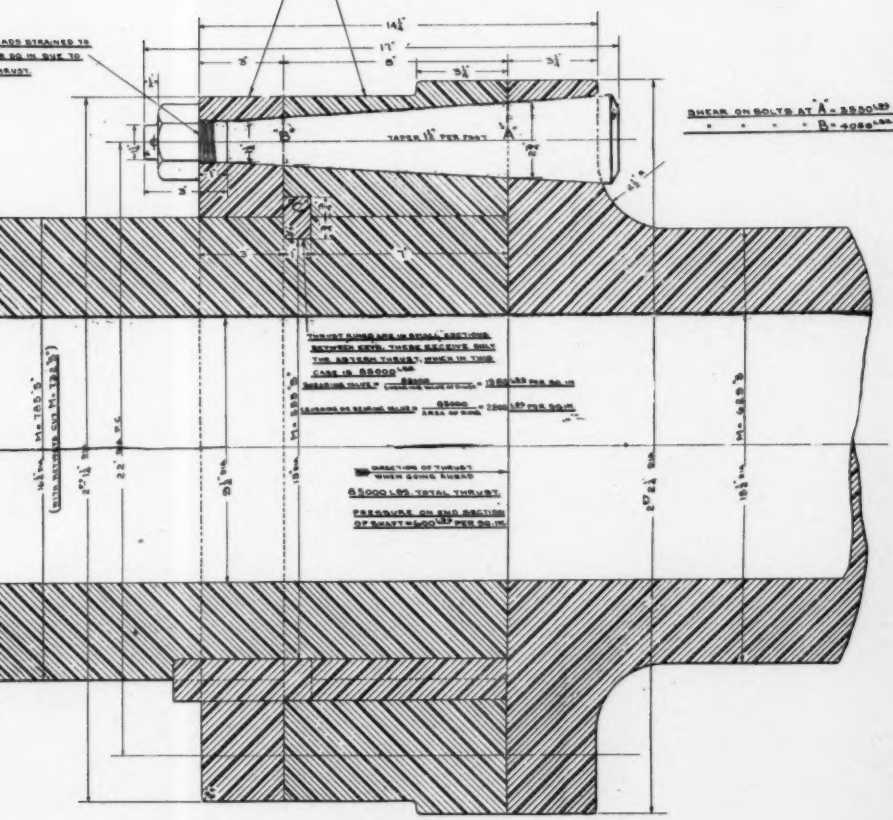
NOTE.- BOTH THESE RINGS ARE MADE SOLID AND FORCED ON SHAFT.
SHAFT IS TAPERED ABOUT ¼" PER FOOT TO INSURE A GOOD FIT AT THE COUPLING.

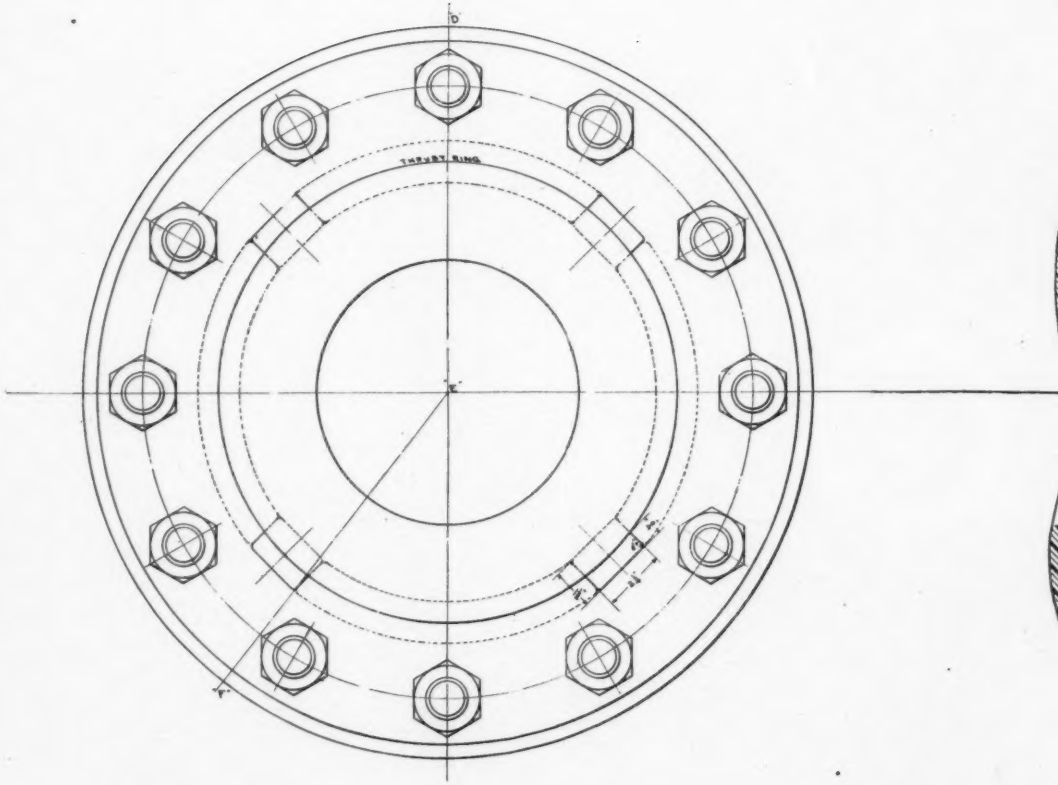


SHEAR ON BOLTS AT "A" = 8930 LBS PER SQ. IN.
" " " " "B" = 4080 LBS PER SQ. IN.

SECTION ON D-E-F

4.- THE RINGS "C" ARE MADE IN SMALL SECTIONS AND ARE EASILY HANDLED, THEY FIT BETWEEN THE KEYS ONLY, AND ARE PROVIDED WITH LIBERAL BEARING SURFACES FOR TAKING THE THRUST WHEN GOING ASTERN.

THIS COUPLING COMBINES THE FEATURES OF EXTREME SIMPLICITY AND THE GREATEST POSSIBLE REDUCTION IN WEIGHT CONSISTENT WITH STRENGTH.



1- THIS COUPLING IS SO DESIGNED AS TO UTILIZE THE ENTIRE LENGTH OF THE SAME FOR KEYS, THE PURPOSE OF WHICH ARE TO TRANSMIT THE STRESSES FROM THE ENGINE OR LINE SHAFT TO THE PROPELLER. THE GROOVE IN THE SHAFT IS LOCATED AT A POINT PREDETERMINED BY THE MOMENT OF RESISTANCE OF THE SHAFT AT THIS REDUCED SECTION, AND THE AMOUNT OF WORK TRANSMITTED THROUGH THAT PORTION OF THE KEY BETWEEN THE GROOVE OR RECESS AND THE END OF THE SHAFT IS PROPORTIONAL TO THE STRENGTH OF SHAFT AT SAID RECESS.

2- THAT PORTION OF KEY BEYOND THE RING NEED ONLY BE OF SUFFICIENT LENGTH TO TRANSMIT THE TORSION REQUIRED TO MAKE UP THE FULL TORSIONAL STRENGTH OF THE ADJOINING SHAFT, THIS IN MOST CASES IS THE LINE SHAFT, AND AS SUCH, IT IS ALWAYS MADE OF A LESS DIAMETER THAN THE PROPELLER SHAFT, WHICH ENABLES US TO DESIGN THE COUPLING IN THE MANNER DESCRIBED WITHOUT ADDING ANY ADDITIONAL METAL TO THE INSIDE OF THE SHAFT WHICH IS GROOVED.

3- THE COUPLING BOLTS ARE DESIG
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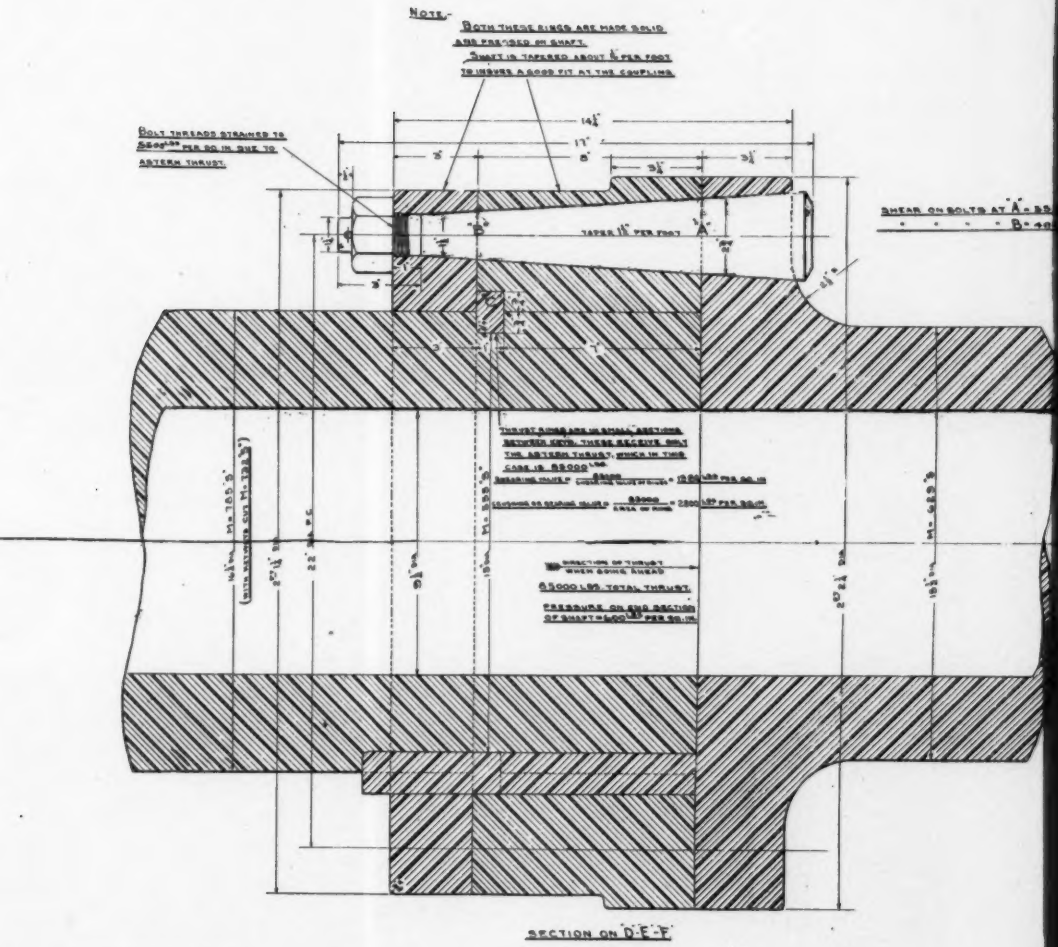
NOTE.
BOTH THESE RINGS ARE MADE SOLID AND PRESSED ON SHAFT.
SHAFT IS TAPERED ABOUT ¼" PER FOOT TO INSURE A GOOD FIT AT THE COUPLING.
BOLT THREADS STRAINED TO 5600 LBS PER SQ. IN. DUE TO ASTERN THRUST.
SHEAR ON BOLTS AT A = 83
B = 49
TAPER 1¼" PER FOOT
THRUST RINGS ARE IN SMALL SECTIONS BETWEEN KEYS. THESE RECEIVE ONLY THE ASTERN THRUST, WHICH IN THIS CASE IS 88000 LBS.
DIRECTION OF THRUST WHEN GOING AHEAD
85000 LBS. TOTAL THRUST.
PRESSURE ON END SECTION OF SHAFT = 600 LBS PER SQ. IN.

SECTION ON D-E-F

...E BOLTS ARE DESIGNED SO AS TO ...E SHEARING LOAD AT THE JOINT "A" ...CH PROPORTIONS OF THE TOTAL LOAD ...MAY BE REQUIRED BY THAT PORTION OF ... RING, THIS MAKES IT ABSOLUTELY ...A TAPER BOLT AND THE SMALL END OF ... BE PROVIDED WITH THREADS OF ... TO WITHSTAND THE THRUST OF THE ...GOING ASTERN WHICH IS VERY SMALL ...TH THE ACTUAL STRENGTH OF THE

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THIS COUPLING COMBINES THE FEATURES OF EXTREME SIMPLICITY AND THE GREATEST POSSIBLE REDUCTION IN WEIGHT CONSISTENT WITH STRENGTH.

2. These keys are fitted from the end of the shaft with the collars in place, so that the keyways may be broached, or otherwise accurately finished.

3. The ring for taking the astern thrust of the propeller is of ample section for the work required (its shearing stress is less than 2,000 pounds per square inch). It is divided into four sections between the keys, and the parts are easily fitted and handled.

4. The location of the thrust ring is such that the bearing surfaces of the keys between it and the end of the shaft are designed to transmit the torsion of the reduced section of the shaft at the ring. The remaining portions of the keys beyond the ring are designed to transmit a torsion equivalent to the reduction made by the cutting of the groove in the shaft.

5. The torsional strength of the coupling is thus equivalent to that of the propeller shaft, with the reduction due to the cutting of four keyways. But these four keyways still leave the strength of coupling in excess of the line shaft to which it is immediately adjacent (see moduli shown on drawing, 732 and 629 respectively). Hence there is no need to increase the strength of the propeller shaft by making the hole smaller at this point.

6. The coupling bolts transmit the whole torsion only at the section "A." Hence the other sections are reduced in proportion. The section at "B" transmits only the torsion due to the keys beyond the ring. The section at the bottom of the threads is only that required to take the astern thrust of the propeller. This makes the tapered bolt ideal for the work required, and makes the fitting easier and weight less.

7. The loose collars on the shaft may be fitted with any desired taper, so they will draw up to a close fit. This also facilitates removal.

8. The end face of the coupling and propeller shaft is perfectly plane, hence the adjacent section of line shaft may be lifted out after removal of coupling bolt.

9. The total weight of the coupling is reduced 1,295 pounds, equal to 2,590 pounds for one ship (see details of weights

shown on sheet 10). The calculated stresses are moderate in all cases, and nearly uniform. The design is the simplest and safest possible, and there are no difficult problems in manufacture.

THE LOVEKIN INBOARD LOOSE COUPLING FOR THE U. S. BATTLESHIP "NEW HAMPSHIRE."

Conditions Required.

Horsepower = 16,500 total, = 8,250 H.P. each shaft.

R.P.M. = 120, = N.

Inch pounds per minute = $8,250 \times 33,000 \times 12$.

Let M represent moment or torque of shaft in inch pounds.

Hence work done by shaft = $2\pi NM$ inch pounds.

Therefore $2\pi M = 8,250 \times 33,000 \times 12$.

$$M = \frac{8,250 \times 33,000 \times 12}{2\pi \times 120} = 4,335,000 \text{ inch pounds.}$$

Line shaft is $15\frac{1}{2}$ inches diameter outside.

Nine inches diameter hole.

We will design coupling in accordance with the strength of the line shaft.

Moment of resistance to torsion of hollow shafts = $.1,965\ s\ \frac{(D^4 - d^4)}{D}$,

Where S = stress in pounds per square inch.

D = outside diameter.

d = inside diameter.

For line shaft $D = 15\frac{1}{2}$ $D^4 = 57,720$

$d = 9\frac{1}{2}$ $d^4 = 8,145$

$D^4 - d^4 = 49,575$

$$M = .1963 \times \frac{49,575}{15.5} \times s = 629\ S.$$

Hence, $S = \frac{4,335,000}{629} = 6,900$ pounds per square inch stress on normal section of line shaft due to torsion.

Thrust calculated previously—

85,000 pounds actual estimated.

To resist this thrust when going astern, fit a circular key in sections broken by the 4 keys put in to transmit torsion.

Make this thrust key 1 inch wide and $\frac{3}{4}$ inch deep each in shaft and collar.

It is cut by 4 torsion keys $2\frac{1}{4}$ inches wide each.

Shearing stress on thrust key:

Shaft $16\frac{1}{2}$ inches diameter — circumference = 51.8 inches.

4 times width of torsion keys $4 \times 2\frac{1}{4} = 9.$

Length of shearing area remaining 42.8 inches.

Key is 1 inch wide, hence shearing area is 42.8 square inches $= \frac{85{,}000}{42.8} = 1{,}980$ pounds per square inch shear, say 2,000 pounds per square inch.

Crushing or bearing value on thrust key in shaft:

Mean diameter of surface in key seat

$= 15\frac{3}{4}$ inches — circumference = 49.5 inches.

Length cut by torsion keys = 9.

Circumference remaining = 40.5

Depth of key in shaft $= \frac{3}{4}$ inch.

Hence, area under pressure $= \frac{3}{4} \times 40.5 = 30.4$ square inches.

Bearing value on circular thrust key $= \frac{85{,}000}{30.4} = 2{,}800$ pounds per square inch torsion:

Consider 4 keys — $2\frac{1}{4}$ inches wide and $\frac{3}{4}$ inch deep each in shaft and collars. Find length of bearing surface required in keys to transmit the torque of 4,335,000 inch pounds.

Consider the pressure on the sides of the key seats concentrated along the middle line of the side of the seat; *i. e.*, at a distance of $\frac{3}{8}$ inch from outside of shaft.

Therefore the arm of this concentrated pressure will be $8\frac{1}{4} - \frac{3}{8} = 7\frac{7}{8}$ inches.

For a torque of 4,335,000 inch pounds the pressure at an arm of $7\frac{7}{8}$ inches $= \frac{4{,}355{,}000}{7.875} = 550{,}000$ pounds for all keys.

This divided between 4 keys gives $\frac{550{,}000}{4} = 137{,}500$ pounds for each key.

Make each key with 10-inch length of bearing surface on sides.

Then bearing value on key seats $\frac{3}{4}$ inch deep $= \frac{137,500}{10'' \times \frac{3}{4}} =$ 18,333 pounds per square inch, say 18,300 pounds per square inch.

Locate the thrust ring at such a point that the key surface between the ring and the end of shaft will be proportionate to the reduced section of the shaft through which this portion of the torsion must be transmitted. The portion of the keys beyond the ring will then be of just sufficient length to transmit the torsion required to make up the full torsional strength of the line shaft.

Consider strength of shaft at reduced section.

$$D = 15 \text{ inches} \quad D^4 = 50,625$$
$$d = 9\tfrac{1}{2} \text{ inches} \quad d^4 = 8,145$$
$$D^4 - d^4 = 42,480$$

$$M = .1963 \times \frac{42,480}{15} \times S = 555 \times S.$$

For the full moment of resistance to torsion in the line shaft $M = 629\ S$.

Hence the proportion of the total torsion which may be transmitted by the keys between the ring and the end of the shaft $= \frac{555\ s}{629\ s} = .883$.

Therefore, in 10-inch length of bearing surface of keys the portion between the ring and end of shaft — 10 inches $\times$ .883 = 8.83 inches.

This will leave 10 — 8.83 = 1.17 inches on the other side of the ring.

Say, $8\frac{3}{4}$ inches of key between ring and end of shaft, and $1\frac{1}{4}$ inches of key outside of thrust ring.

For practical reasons, however, the outside ring would not be made so thin as $1\frac{1}{4}$ inches. Make it 3 inches thick.

Then, the length of key between ring and shaft will be 10 — 3 = 7 inches.

Hence, of the torsion transmitted by the keys, 70 per cent.

will be carried by the key between the ring and end of shaft, while 30 per cent. will be carried by the outside portion of keys and their 3-inch collar.

Find strength of tail shaft with keyways cut out:

$$D = 16\tfrac{1}{2} \text{ inches} - D^4 = 74{,}120$$
$$d = 9\tfrac{1}{2} \text{ inches} - d^4 = 8{,}145$$
$$D^4 - d^4 = 65{,}975$$

For full section $M = .1963 \times \frac{65{,}975}{16.5} s = 785\, s.$

Consider weakening effect of 4 keyways $\frac{3}{4}$ inches deep, $2\frac{1}{4}$ inches wide.

Cross-sectional area of each keyway $= \frac{3}{4} \times 2\frac{1}{4} = \frac{27}{16}$ square inches.

For approximation assume center of gravity of section removed is at $\frac{3}{8}$ inch from circumference.

Hence, mean radius of cross section from center of shaft $= 8\frac{1}{4}$ inches $- \frac{3}{8} = 7\frac{7}{8}$ inches.

Allow a shearing or torsional stress of S pounds per square inch.

Then, moment of 4 keyways removed $=$

$$4 \times 7\tfrac{7}{8} \times \frac{27}{16} \times S = 53.1\, S.$$

Therefore, the net moment of resistance to torsion in tail shaft where reduced by the cutting of keyways will be $785\, S - 53.1\, S$, showing that there is still a margin over the strength of the line shaft, which is $629\, S$.

SHEAR OF BOLTS IN INBOARD LOOSE COUPLING.

Use 12 bolts.

Make pitch circle 22 inches $=$ 11-inch radius.

For a torque of 4,335,000 inch pounds the pressure at 11 inches radius $= \frac{4{,}335{,}000}{11} = 394{,}100$ pounds.

This divided between 12 bolts $= \frac{394{,}100}{12} = 32{,}842$, say 32,850 pounds each bolt.

Allow 5,550 pounds per square inch shearing stress, as in previous arrangement.

Then area of each bolt $= \frac{32,850}{5,550} = 5.9$ square inches.

$2\frac{3}{4}$ inches diameter gives 5.94 square inches area to withstand total shear.

For 30 per cent. of shear area of each bolt $= .3 \times 5.9 =$ 1.77 square inches.

$1\frac{1}{2}$-inch diameter gives 1.767 area.

For tension due to thrust of 85,000 pounds.

Tension on each bolt $= \frac{85,000}{12} = 7,100$ pounds.

With $1\frac{1}{2}$-inch bolt area at root $= 1.295$.

Hence, stress per square inch $= \frac{7,100}{1.3} = 5,500$ pounds

reduction in weight due to change in form of inboard loose coupling.

COMPARISON OF WEIGHTS OF OUTBOARD AND INBOARD COUPLINGS AS USED ON U. S. BATTLESHIPS *KANSAS* AND *NEW HAMPSHIRE*.

For one shaft.	*Kansas*, fin.	*New Hampshire*, fin.
Outboard coupling complete with keys and collars ; or as in *New Hampshire*, with bolts and lock ring, pounds	3,791	2,759
Shaft ends in same, pounds	2,788	2,400
Kansas coupling being $15\frac{1}{4}$ inches longer than that of *New Hampshire*, add weight of that much $16\frac{1}{2}$-inch shaft to *New Hampshire* weight, pounds	.....	721
	6,579	5,880
Inboard couplings, sleeves, pressure rings, bolt keys, pounds	2,201	1,174
Weight of shaft end in same, also weight of entire flange on line-shaft end, pounds	1,376	1,004
Kansas coupling being $2\frac{1}{4}$ inches longer than that of *New Hampshire*, add weight of that much $16\frac{1}{2}$-inch shaft to *New Hampshire* weights, pounds	.....	104
	3,577	2,282
	3,577	*Kansas.*
	2,282	*New Hampshire.*
Saving in weight	1,295	pounds each.
	2	couplings.
Total saving in weight	2,590	for two couplings

SUGGESTIONS FOR THE CARE AND OPERATION OF NAVAL MACHINERY IN THE ENGINEER DEPARTMENT, U. S. NAVY.

By H. C. Dinger, Lieutenant, U. S. N., Member.

PART II.

GENERAL RULES FOR THE OVERHAULING OF MACHINERY.

In making repairs the most important ones should be begun first, and due consideration should be given to the different jobs. The allotment of work should be made so that there is the least interference between the various gangs of mechanics.

The repairs to make first should be those designed to get the machinery of the vessel into efficient steaming condition, and to this end the necessary adjustment of bearings should be made first. If there are no special repairs the whole department should be thoroughly cleaned.

The following is a list of ordinary repairs generally necessary.

Adjust all loose bearings.

Remake all leaky joints; overhaul and grind in all leaky valves; open and examine cylinders and valve chests of main engines.

Examine and overhaul valves of air and feed pumps, water end.

Clean bilges.

Clean and overhaul wicks, oil cups and oil piping.

Open and examine main condensers; test for tightness; boil out condensers if tubes show grease on them.

Clean feed and filter tanks, and renew filtering material.

Scale evaporators.

CYLINDERS.

The cylinders are opened for the purpose of examining the interior surfaces, also to examine piston rings and springs; to test the tightness of piston-rod nut and follower bolts; also to wipe out moisture and put on a coat of vaseline for lubrication and to protect the surfaces of the cylinders from rust while engine is standing idle.

In opening cylinder care should be taken not to destroy the gasket, which should have been thickly covered with graphite to ensure an easy parting. The cylinder should first be wiped out with kerosene or turpentine in order to remove the moisture and any rust that may have formed. After this the surface can be examined.

Examination.—Any scores or flats should be observed and a record of them made, so that any further increase in such defects may be known. The follower should be lifted occasionally; rings cleaned, springs examined and broken ones replaced.

Points to be noticed in examination are: whether the surface of cylinder is smooth and bright all over, whether the wear is on one side (this can be told by observing the appearance of surface before the cylinder is wiped out). If one-sided wear is observed the piston is probably out of line. The surface of piston ring should be smooth and bright all over. If rod is bent or piston not working in line the ring may wear wedge shaped in cross section. The edges of ring may become very sharp and in such condition may at times nip the cylinder walls. It may often be advisable to round off slightly both the edges of the rings and those of the pistons and followers. The ends of the split rings should be fitted with efficient keepers to prevent them from vibrating.

The prominence of the shoulder at the end of stroke on pistons that do not overrun the counterbore should be noticed and, if of any amount, the shoulder should be scraped down. With pistons that overrun the counterbore, as is the case on all late vessels, there will, of course, be no shoulder.

If the cylinder liner is badly scored or worn it may have to be rebored or a new one fitted.

Scoring.—Scoring of cylinders is caused by (1) Hard particles of dirt or grit left in the posts or in the cylinder itself, or brought from the boilers through priming, or from particles of rust that have scaled off from the surfaces. (2) By the edges or ends of rings nipping cylinder walls; this may be caused by countersunk bolts not being perfectly flush with the surface. (3) Packing ring being pressed out too hard against the cylinder walls by the steam pressure. To avoid this solid rings, or rings whose extension is limited, are employed.

A one-sided wear or flat is caused by the piston being out of line. If the piston has considerable clearance in the cylinder it is likely to be shoved from one side to the other by the action of the steam, and this also may cause local wear. The breaking of several springs in one place may cause the piston to be shoved over to this side so that the edge of follower or piston will wear on the cylinder.

PISTON RINGS.

The purpose of the piston ring or rings is to maintain a steamtight joint between the piston and the walls of the cylinder. This steamtightness must at the same time be maintained without excessive friction, which would cause wear on the walls of the cylinder.

The following are the requirements for piston rings from the machinery specifications of the armored cruisers *Montana* and *South Carolina.*

"Each piston will have a packing ring one inch thick and two inches wide on the bearing surface, cut obliquely and tongued. The play of the ring will be limited by an adjustable distance piece, lug and clamp, or some other suitable device.

"The packing ring will be set out by steel springs of approved pattern, all set to an equal and proper tension. The springs will be so secured in the piston as to be firmly held in place and easily inserted or removed."

The pistons of the auxiliaries generally use cast-iron snap rings cut diagonally. If these rings and the cylinder walls have a good, bright surface all over, it shows that they are

properly tight. If the surface is not bright in spots the ring is loose and a new one should be fitted. For small engines new rings can be made on board from castings kept for the purpose. The ring is turned slightly larger than the diameter of the cylinder and is then cut diagonally in one place and then sprung into the piston, the natural tendency is for the ring to spring out to its former shape, and it thus presses against the cylinder walls. Special care must be taken that the edges of the ring make a good, tight fit with the flange of the piston. The principal trouble encountered with these rings is the breaking off of the ends.

Large pistons are fitted with followers. To examine the ring the follower is lifted. The follower holds the packing ring in place. The follower is secured by steel studs screwed into the piston, the bodies of the studs are usually square and pass through square holes in the follower. The studs are made square in the body where they pass through the follower so as to prevent their turning, and thus coming out. The nuts are each secured by a brass or steel split pin or else all the nuts are wired together. In overhauling, special precautions must be taken to see that the nuts are tightly secured and all split pins in and their ends properly flared.

Piston Springs.—The type of piston spring usually used in the Navy is the ordinary steel coach spring. These springs are inserted in the space between the rings and the body of the piston. They press the rings out against the cylinder wall and make a steamtight joint. They also serve to center the ring over the piston, and in this way tend to guide the piston centrally through the cylinder. Springs occasionally break and have to be renewed.

For low-pressure cylinders the piston rings are generally made in sections and double, one ring resting on the other. The sections are arranged so that the parts of the two rings break joints with each other. The springs behind the rings are depended upon to give tightness. It would be somewhat difficult to get single rings, in one piece, for the large low-pressure cylinders, hence the use of sectional rings. (Fig. 12.)

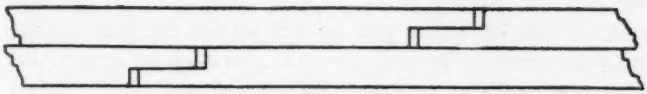
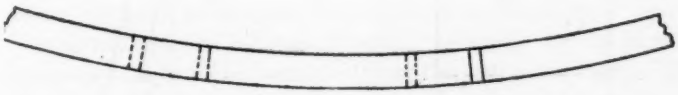


Fig. 12.



For small engines, and for the intermediate cylinders of triple-expansion engines, ordinary split cast-iron rings are generally used. To prevent the ends of rings breaking off, due to vibrations, they are held to each other by various styles of keepers which, in holding the ends together, check the vibrations. Tongue pieces are fitted between the ends to prevent the passage of steam. (Fig. 13.)

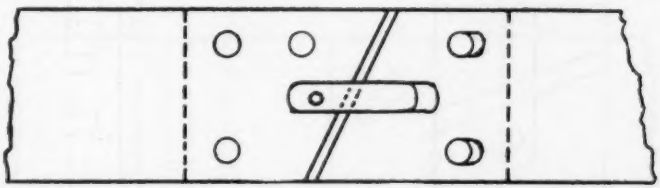
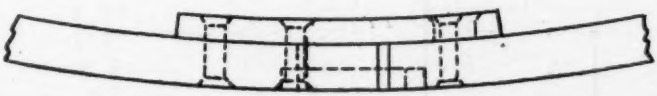


Fig. 13.



For high-pressure cylinders and for intermediate-pressure cylinders using very high-pressure steam, many difficulties

have been encountered with packing rings. With the ordinary split ring the steam gets in behind the ring and presses it out hard against the cylinder walls to such an extent that the cylinder walls are cut and scored and the rings broken. When double rings are used there is also the liability of the steam coming through the joint between the two rings.

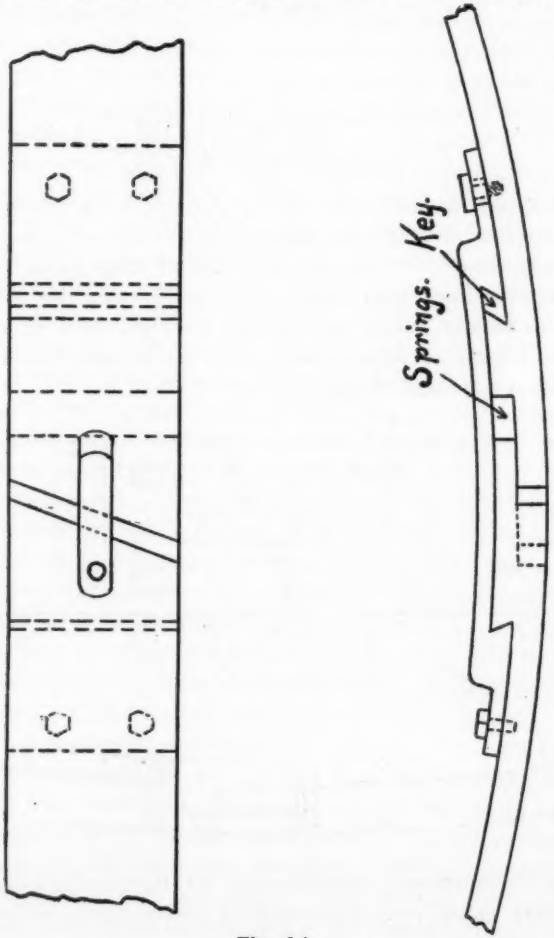


Fig. 14.

To prevent these troubles spring rings fitted with a device for limiting their extension are used. (Fig. 14.) This ring is adjusted so that it can open to a fit in the cylinder, but no further, and can close within certain limits. When the ring wears sufficiently to cause leakage additional extension can be given by filing out on the clearance spaces or lugs or keys till ring is sufficiently enlarged.

This type of ring sometimes gives trouble by reason of bolts, screws or other parts coming loose or breaking.

Another practice is to use a solid ring bored out to fit the cylinder snugly when warm. With this type the danger of ring expanding too far, bolts getting loose, or leakage past joint is entirely avoided. These rings, when carefully fitted to an accurately bored cylinder, give good satisfaction. There is naturally very little wear since there is no pressure forcing the ring against the cylinder. A new ring can be fitted when sufficient wear to warrant it has taken place.

To enable an adjustment for wear to be made and at the same time secure the advantages of the solid ring, the ring can be cast solid with lugs in one place. Then cut the ring square

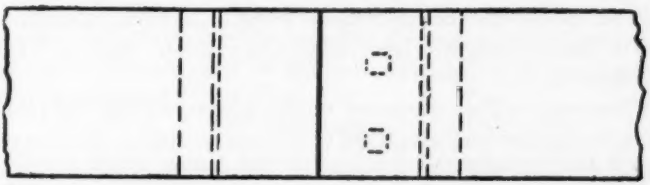
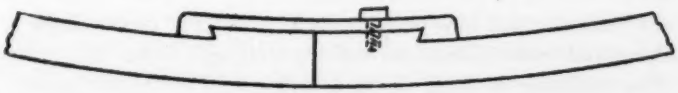


Fig. 15.



at the lugs and fit a clamp over the lugs to hold the ends tightly together. When wear takes place the lug or clamp can be shaved off slightly and a liner inserted between the ends to make up the necessary distance, and the lug replaced. (Fig. 15.)

Springs are fitted behind the solid ring in order to keep the piston centered and to help guide the piston through the cylinder.

It is of the utmost importance to have the joint between the edges of ring and flanges of piston an absolutely tight fit. To ensure these surfaces being true the edges of rings may have to be scraped to a fit, and the follower may have to be lined up so that it will allow exactly enough space for the ring. The follower should not be brought down enough to grip the ring, the ring should be free to move on the application of pressure.

In adjusting rings for tightness it will be safe to get the ring so that it is a tight fit in the cylinder, but at the same time so that it can be moved around without great difficulty. When the engine is warmed the cylinder liner should expand about the same amount that the ring does, so if it is tight when cold it should also be tight when warm. If on examination the surface of ring is bright all around it shows that the ring is a tight fit.

Clearance.—The clearance of the piston, top and bottom, must be verified and measured from time to time. With continual use, on account of wear at crossheads, crank-pin and main bearings, the piston drops down. Marks showing the clearance should be stamped on crosshead slide; these should be verified occasionally by disconnecting piston rod, allowing it to touch at top and bottom and observing whether the positions correspond with the marks. The distance from mark when piston is on striking point to mark when on center and connected up, is the linear clearance for that end. The actual clearance can also be measured by putting pieces of putty above and below piston and moving the engine over the center. The thickness of these pieces of putty will be the clearance.

The dropping down of piston may be rectified by placing liners under piston or under the end of connecting rod sufficient to raise it up the necessary distance.

Testing for Tightness.—The tightness of piston may be observed by passing a light around the edge of ring and at the same time jacking piston down, with link central; where the flame is caused to flicker air is leaking through, and at this point the piston is not tight. The ease or difficulty with which an engine can be moved with links in center will give a general indication of the degree of tightness of pistons and valves.

To test tightness of piston under steam, put engine in position so that steam is admitted to top of piston, then open lower indicator cock. If steam is now admitted and it is found to issue through indicator cock, it shows that the piston is leaking.

Closing Cylinders.—Everything should be replaced in proper shape and carefully and tightly secured, special precaution should be taken to see that follower nuts are tight and that all split pins are in place with ends properly flared. Great care should be taken that no tools, waste or material of any kind is allowed to remain in the cylinder. A special inspection should be made just before putting on the cylinder head. Before closing up, the cylinder should be coated with vaseline mixed with graphite. The vaseline preserves the surface from rust while the graphite serves as a lubricant and develops a good wearing surface.

VALVE CHESTS.

Valve chests should be overhauled at the same time that the cylinders are, so that one job can be made of the whole thing. The surfaces of the valve chests are subject to the same general treatment and care that the cylinders are. It is extremely important that the surfaces of packing rings and the wearing surfaces of liners be bright, smooth and of a tight fit. Graphite should be liberally supplied to the working surfaces so that a good wearing surface may be developed.

Special care should be taken that the valve-rod nuts are tight. The wear of valves should be examined periodically. Where piston valves are fitted with solid rings, split and secured

with clamps or bolts, and arranged so that they can be extended as wear takes place, liners to make up the wear and make ring tight should be fitted when necessary.

The surfaces of slide valve must be kept in good condition in order to avoid excessive friction and steam leakage. Whenever scores or rough places are discovered they should be smoothed. Small slide valves can be brought to a true surface by means of the surface plate, and the valves themselves used to bring the faces to a fit.

Valve Balancing Devices.—In order to prevent the weight of the valve coming upon the valve gear and eccentrics, various devices for taking up this weight are fitted.

Balance Pistons.—These work in a small cylinder cast in the upper bonnet of the valve chest. The balance piston is secured to an extension of the valve stem. The part below the piston is in communication with the valve chest and the part above the piston is piped to the condenser.

The balance piston is designed so that the difference in pressure on the two sides of the piston will be sufficient to take the weight of the valve and its connections.

This balancing is sometimes effected by varying the diameter of the two ends of the valve; the excess pressure on the larger end balancing the weight of the valve and its connections.

Balance pistons are fitted on slide valves as well as on piston valves.

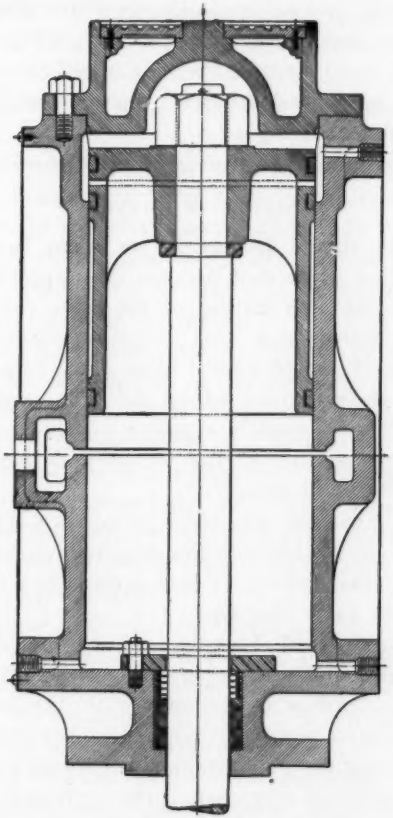
Slide valves are also often fitted with relief rings on their backs, but these relief rings are fitted to take the pressure of the valve off the valve seat.

Assistant Cylinders.—These are sometimes fitted for taking up the inertia as well as the weight of the valve and valve gear.

Both Thom and Joy's assistant cylinders have been used in England extensively.

The Lovekin improved assistant cylinder is an improvement over both the Joy cylinder and the Thom momentum cylinder; it is used on a number of American merchant vessels and is fitted over each valve on the engines of the U. S. S.

Washington, *Kansas* and *New Hampshire*, there being fourteen cylinders for the two engines in each vessel.



LOVEKIN ASSISTANT CYLINDER.

A cut of the arrangement of this device is shown. The annular space at center is connected to some one of the receivers of the engine which will give the desired pressure for operation. Sometimes this steam taken from the H.P. chest is controlled by a differential regulating valve which gives a pressure suitable for any speed at which the engine may be running. By inspecting the sketch it will be seen that there

is a cushioning effect at each end of travel of valve, and that the valve is assisted in movement by the expansion of the cushioned steam and admission of receiver steam at about middle of down stroke and near end of up stroke. The object of the assistant cylinder is to take the weight and the inertia forces of the valve and gear off the crank shaft, eccentrics, links and other parts of the valve gear.

Assistant cylinders can readily be fitted to old engines.

SETTING OF VALVES.

Opportunity should frequently be taken to examine the setting of valves to see that they are not improperly set. An indication of improper setting of the valve can be observed from the indicator card; this is perhaps the best guide in this matter, as it would appear to be a safe rule that as long as the cards show nothing wrong the valve should not be altered. Engines are usually supplied with steel laths marked with the proper setting of the valves and these are used to verify the position of valves.

In large engines the lap and lead for each end of cylinder are carefully calculated from the original design and measured from the actual setting. These figures are recorded on the drawing of the valve diagram.

To measure the lead of a valve, put engine on center with link in full gear and measure opening of steam port for that end. If this distance corresponds with the lead given on drawing, the valve is set as designed. One end will ordinarily be sufficient for verification, since if lead changes at one end the other should also change the corresponding amount. It is, however, better to measure both ends. Large engines are fitted with peep holes covered with bolted plates, through which the setting of valves can be observed.

Valves Dropping.—In the course of time, due to wear on their collars, valves will drop, thus decreasing the lead at one end, increasing it at the other. If valve takes steam on the outside, a dropping down of valve will increase the lead on top and decrease it at bottom.

On piston valves not fitted with peep holes, taking steam on the inside, it may be impossible to get at the port to measure the opening; in such cases the lead may have to be determined by noting the position of valve by measurement from some convenient point on valve chest, and then, by taking the actual distance, from drawings, the lead may be calculated.

To bring valve up into proper position, distance pieces may be put above shoulder of valve stem, or an easier way is to place liners under the valve stem. This takes into account the dropping down of valve due to wear, and hence the shortening of valve stem. Valves may also be deranged by shifting of the nuts securing the valve.

With smaller auxiliaries, many of which have neither lap nor lead for the valves, and of the setting, of which there are probably no records, the following rough-and-ready method suffices: The engine is put on a center, the valve placed approximately and the lead measured. The engine is then put on the other center and the lead again measured. The valve is then shifted till the leads are equal. On vertical engines allow slightly more lead on bottom than on top to allow for wear and weight of parts.

To measure the leads of valves a long, thin wooden wedge is often used. The wedge is inserted in the opening as far as it will go, the edges of opening will leave a mark on the wedge which is a measure of the lead. When the marks left by either end are alike the leads are alike.

On auxiliary engines having two cylinders working on same shaft special care must be taken that the eccentrics are connected up correctly. It sometimes happens that one cylinder is connected up for backing when the other is for going ahead. When this happens the engine will run neither way.

Engine on Center.—To put an engine on center, or rather an engine crank on dead center, proceed as follows: Put turning engine in gear, move engine so that crosshead is near end of stroke, scribe a mark across crosshead and guide for

this position; call this mark A. At the same time scribe mark across shaft and its adjacent brass, or coupling and some adjacent fixed part; call this B, B′ on brass. Then continue to turn engine until the mark A again coincides, then mark adjacent brass at point where B now is; call this C. Now find the center between B′ and C on brass; call this D. Bring B and D together and the engine will be on the center.

When engine is built, marks are usually put on some coupling or on the turning wheel, which when trammed from some fixed point indicate the dead center of the various cranks. Such marks should, however, be verified from time to time.

Care of Valve Gear.—The overhaul of that part of valve gear outside of valve chest consists of adjusting eccentric straps, link blocks and gibs. The eccentric straps are usually adjusted so that they can be moved by hand; on large engines "leads" are taken. On cylinder having two piston valves, yokes and guides are fitted; these guides require lining up from time to time.

Other matters in regard to valve gear are treated under the heading of lining up engines.

BEARINGS.

The overhaul of loose bearings is one of the chief items in the work of repairing on board ship. The bearings that require the most attention are the crosshead and crank-pin bearings.

Removing Parts.—Facilities for handling are of great value in expediting the work of overhaul. The fitting of hooks and eyebolts for hooking jiggers and pulleys, and tapping brasses and other parts for screws and bolts, is of the utmost importance, and these facilities for handling parts can be continually added to by the ship's force.

Crank-Pin Brasses.—Eyebolts are screwed into the upper ends of crank-pin bolts and the ends of chain falls hooked into eyebolts; the nuts are then slacked back till bolts get clear; the brass is then dropped down and swung clear. The crosshead is secured by hanging it, or shoring it up. The crank can now be turned till the upper brass is clear, and that brass

can then be removed. To facilitate the examination of the upper brass this may be held fast to the end of the connecting rod by small links and set screws at each side. The crosshead with connecting rod attached is then blocked up, usually done by putting strongback across crosshead guide. The crank is then jacked away from brass so bearing can be observed from below.

Crosshead.—To strip crosshead it must first be hung up ; this is usually done by a bolt being screwed into a hole in slide below crosshead. Other times it is done by shoring or blocking it up. Some provision in the way of blocking must be made on which to rest the upper end of connecting rod when free from crosshead, and this must be fitted so that it can slide down to some extent to permit an examination of the bearing to be made. For large engines tackles and eyebolts are necessary to handle brasses. A connecting rod may also be hung up by removing bullseye in cylinder head and attaching a supporting tackle to upper end of piston rod.

Main Bearings.—The top brasses need simply to be lifted, the bottom brasses are made so that they can be rotated till they come clear. For large engines a bar having a projection that will engage the top face of lower brass is secured to crank web. When the crank is rotated by turning engine the bar moves the lower brass around until it can be taken away.

Examination of Brass.—Examination should be made to see the condition of journal—whether scored or smooth, whether truly cylindrical or whether it has hollows or humps. The presence of these irregularities can be told by eye and touch, but to be accurate the journals can be calipered. The white metal of brass should be smooth, clean and sound, the oilways clear.

Oilways.—A very important item is the character of the oilways. Oilways should be cut so that the oil is brought to the point of greatest pressure, and they should not be cut so that the oil tends to run out at end of journal. The figure-of-eight oil groove is generally the most satisfactory for small

bearings. Oilways should not be cut in line with the axis of journal, since this may weaken the brass very materially in one place. Oilways and oil holes should not be made too large, since too much oil will collect in one place and this tends to wear a hump.

Readjustment.—Bearings must be tight so that there is little play, but there must also be a certain amount of play to allow a layer of lubricant to keep metal off metal. To reduce this play as much as possible and at the same time prevent heating is the result to be arrived at. The amount of play must be uniform, because if it is not, only a small portion of the bearing is a real bearing surface, and the friction on this limited bearing surface will be excessive. The bearing should therefore be perfectly smooth, the play uniform over the whole surface and as small as possible while still allowing for a layer of lubricant.

Bearings may only require adjustment, that is a tightening up on the brasses. The amount of play allowed will depend on: (1) The size of bearing. (2) The general alignment. (3) The speed of revolution. (4) Whether the motion is vibratory or revolving. The amount of play is usually expressed by the numbers of the standard wire gauge.

Large bearings, such as main and crank-pin bearings, are set up to 27 to 32 lead, American gauge; small bearings can be set up as tight as 36, but with this very nice adjustment great accuracy in fitting is necessary.

The clearance in the bearing is measured by taking leads. The upper brass is taken off, a thin lead wire is laid on journal either as in 1 or 2, 2 is perhaps better. (Fig. 16.)

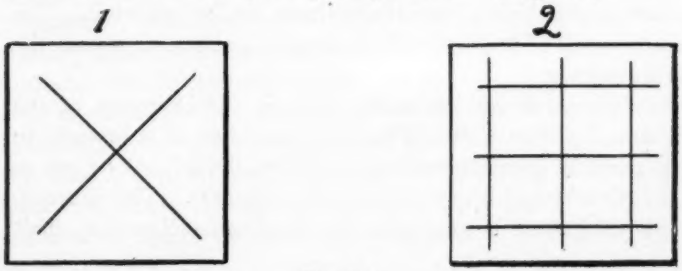


Fig. 16.

The upper brass is put on and the nuts set up and their position marked. The brass is then taken off; the thickness of the wire shows the clearance at that particular point of the bearing, and, by the way in which the wire has been spread out, the general uniformity of the clearance can be seen. If the lead is thick on one side of journal and thin on the other a liner will have to be taken out on one side. If the clearance is too little a thin liner is inserted; if too much, liners are taken out. If the lead shows a general irregularity the brass will require scraping and fitting.

Another method of adjusting brasses is to take out all liners, set brasses up hard, and mark nuts; then slack back nuts the necessary amount for clearance, using the fraction of the pitch of nut to determine the exact amount. Liners are then inserted to the required amount. This method is, however, not very general. When the brasses are carefully fitted and where great care in adjustment can be ensured this is an excellent method. This method is also used where time is limited and rapid adjustment or slacking back necessary, as in case of heating, judgment being used to determine the amount to slack back nuts.

Liners.—Copper, brass, paper and tin are used for liners. Tin is more or less objectionable owing to its liability to rust; it should not be used where there is likely to be salt water.

Fitting Bearings.—Usually the only fitting is smoothing down or scraping scores or rough places.

If the journal is scored or uneven this must first be remedied. Usually a fine file is used to dress the journal, and after this the journal is lapped with an oil stone or with oil-stone powder. Carborundum is sometimes used in dressing journals.

Fitting Brasses.—The bearing surface of brass is smoothed and scraped or bored to an approximate fit. If bored in a lathe it should be bored with sufficient liners in to allow for fitting. To fit the brass the journal is coated thinly with red lead and oil, black lead and oil, lampblack and oil, or putz pomade. The brass is then laid on and moved slightly. Where

the coating adheres the surfaces touch, and these spots have to be scraped. This operation is repeated until the spots are many and uniformly distributed all over the bearing surface. To insure accurate fitting the brasses should be tested in this way when in proper position with bolts in, as by reason of the restraint due to bolts the actual position of the brass may not be the same when bolts are not in.

Brasses should not be reduced all from one side, especially is this so on shaft brasses. It must be remembered that both the upper and the lower brass wear. The bottom brasses of crank-shaft journals must be brought up into alignment by putting liners underneath as wear takes place.

Brasses should be eased off at the sides to prevent nipping of journal when they become warm. This is usually accomplished by distance pieces and by cutting out the white metal for a short distance from edge of brass. The space formed by easing also forms a reservoir for holding the oil—a desirable feature.

Small Brasses.—The degree of tightness and fit of bearing for small brasses can be told by the appearance of bearing and by touch. By feeling how a bearing turns the experienced hand can tell whether it is loose or not. In general, a bearing should be able to be turned without undue effort by the pressure of the hand; an eccentric strap by setting up till the rod will just fall of its own weight. On bearings of any size it is always best to take leads, but for a very large number of small bearings this is not necessary.

Overhauling and Renewing White Metal.—If the white metal is rough or scored it should be scraped smooth. If there are holes in the white metal or the metal is spongy in places the spots may be patched. To do this the holes are thoroughly cleaned out and all bad metal removed, the holes are made a little larger at bottom and washed out with hydrochloric acid. A quantity of white metal is then melted, bearing warmed up and metal poured into holes and allowed to come above the rest of the surface. The new metal should then be hammered into place with pene hammer and, when cold, the surface

should be scraped down to a bearing. Solder or tin can be used in an emergency when no white metal is available.

To Renew White Metal.—If the white metal needs renewing due to cracks, running or thinness of metal, the old metal should be taken out. This can be done by heating in a clean coke fire to melt it out. After the old metal is removed the surface should be cleaned with hydrochloric acid and then tinned. The bottom brass should be placed in position a little lower than when connected up, a piece of putty or clay should be used to close up the ends of the bearing. The white metal can then be poured in ; pour only from one side to be sure to drive out all air. When metal is nearly hard take out brass and hammer surface all over with a pene hammer to make it solid and fill all holes. The upper brass is treated in the same way. Cast-iron mandrels for all large bearings are now generally supplied, and it is more advantageous to use them since a better position and better facilities for pouring the metal can be obtained. When a mandrel is used the brasses can be bolted together, the mandrel set on end and centered in the brasses, and the metal poured into the annular space from the top. Where no mandrels are supplied they can be made by turning up a piece of cast iron to the dimension of the journal. Pieces kept on board for making piston rings will generally answer. After the white metal has been hammered the brass should be bored in a lathe and then scraped and fitted and oilways cut. A spare brass for principal journals, all ready to fit, should be kept on hand so that in case of accident there may be no great delay in renewing one.

Thrust Bearing.—In the course of time the surface of thrust collars wear and give more play than is desirable, the result being that the crank shaft is moved forward. To adjust for this wear the collars, if of horseshoe type, are moved the necessary amount by means of the adjusting nuts provided for each collar.

If the bearing is of box type the whole bearing can be moved by means of the adjusting screws or wedges provided.

When the horseshoes of thrust bearing have worn to such a degree so as to have any considerable play between collars a great strain is brought on engines in backing. It is, therefore, advisable to renew white-metal lining and refit horseshoes in the course of every few years' running.

CROSSHEAD GUIDES.

As the result of wear the crosshead may get out of line, the result being that the piston rod is forced over to one side and a cross breaking stress brought upon the rod at each stroke. The crosshead is brought back into proper position by putting liners under the slipper until the necessary distance is obtained. To determine this disconnect connecting rod and move piston to top of stroke; observe whether there is any clearance between the slipper and the guide. If there is, measure it with a wedge gauge, then let piston drop to bottom of stroke and let crosshead go free. Again take the clearance. If the clearance is the same in each case the slide is in line and the clearance can be taken up by putting a liner under shoe. If, however, the clearance is more at one end than at the other the slide is not in line, but can be adjusted by moving one end out till the crosshead bears evenly all along. The presence of too much clearance can be told without disconnecting anything by inserting wedge gauge between slipper and guide. The backing guide will wear very little, so in adjusting, it is the go-ahead side that will require the attention.

LINING UP ENGINES.

The base of reference for lining up an engine is the center line of shaft. All other parts must be lined up from this as a basis.

All bearings on the crank shaft must be so that their centers are in one straight line from center of propeller hub to center of inner end of crank shaft. The piston rods and pistons must work in a line perpendicular to the line of shafting.

These are the general conditions that must be fulfilled, and in erecting the engine great care must be taken to secure the

proper adjustment and fitting of all the various parts in order to avoid the racking and parting strains that may be brought on various parts when they are out of line. The general tendency due to lack of alignment is a loosening of joints and dislocation and distortion of moving parts.

On board ship the adjustments will consist of relining shafts and crossheads, as they are the points where most wear takes place. The minor attachments, such as valve gear, pump levers, reverse shafts, etc., will also require attention from time to time.

Indication of Engine being Out of Line.—Indications that shaft is out of line are extra pounding, apparent up-and-down movement of any of the bearings, observed by watching the change of clearance between shaft and bearing for different positions of crank.

If two sections of the shaft are uncoupled and the flanges do not meet fair it shows that they are out of line. This can be told very accurately by placing pieces of tissue paper at top and at bottom and at each side of coupling and butting shafts together. If the papers grip all around, the shafts are in line; if they grip in one place and not in another, the shafts are not in line.

Indications of crosshead guide being out of line are knocks in crosshead and a non-uniform clearance between slipper and guide. This change in clearance can be found out as explained under crossheads. The proper alignment of connecting rod to piston rod and crank can be told by observing whether there is any tendency for lower end to move in a fore-and-aft direction on turning of crank, or whether lower end is free, by observing the clearance between crank webs and crank-pin brasses.

In general the lack of alignment of the minor parts of engine are determined by watching for undesired movement, which can be told by touch, observation and measuring in various positions.

LINING UP.

To line up an engine the center line of shaft should first be determined.

The shaft being out and engine stripped of piston, connecting rod and crosshead, crosshead guides and lower brasses of main bearings in place, and cylinders opened, proceed as follows:

A line is run in the position of center line of shaft. In order to obtain this line two points of origin must be found. If the whole shafting is to be relined the two points are the center of stern bearing and the forward end of crank shaft. Usually only a portion of the shaft is relined at one time. When engine is installed wearing gauges for shafts are provided; by applying these the amount that the shaft is low at certain points can be determined. The center of shaft is also marked on parts of engine seating. From these two guides two points of the center line can be found and a cord or wire stretched between these. Such points must, however, not be far apart, as there is considerable sag in the cord or wire. Points along the shaft may also be determined by lines of sight, and points along the line marked on wood battens.

The center line of shaft being determined, the exact distance down to the bearing surface of lower bearings can be calipered at points along the bearing and they can be raised or lowered till all are in line. A line of sight along the bearings may be of help in bringing them into position. Next, the middle line of all bearings must be brought into the vertical plane of the center line of shaft. This can be done by marking the middle of each brass and then moving brasses sideways till all are in position. When this has been done the main bearings should be in line. By using a straightedge between each two bearings further discrepancies may be located and rectified. Small discrepancies are rectified either by fitting liners under the bearing or by scraping the surface of bearings at high points. On small engines, and often also on a large one, the shaft itself can be used to fit the bearings. The journals, having been smoothed up, are given a light coat of red lead, then shaft is put in place and moved slightly. The red lead will adhere on the high spots, and these can then be scraped, and the operation repeated till a good fitting line of bearings is obtained.

Lining up Cylinders.—The center line of cylinder must be perpendicular to the line of shaft and in the same vertical plane. The intersection of the line of the cylinder and the line of shaft is obtained approximately by measurement on bearing. The center of upper counterbore of cylinder is taken as the upper point of origin. A board having a hole in it is secured over center of cylinder so that the hole is over the center. A line or wire with a wooden toggle at end is then passed through this hole; the end of line is made fast below so as to come at the point of intersection of line of cylinder and shaft. The upper end of line is then moved till it comes exactly in the center of the counterbore. This may be done by calipering, or by the use of a wooden stick, slightly shorter than the radius of counterbore, having a pin stuck in its end. This pin can be pushed out till the exact radius of counterbore is obtained, and then by measuring around on all sides the line may be exactly centered. The upper end of line having been centered, the lower end must be brought into exact position. First, it must be moved so that it comes just in contact with the shaft line. Secondly, it must be at right angles to the center line of shaft. To see whether it is at right angles a large square may be used. Rest the square with one edge on the main bearings, which have previously been lined up; bring the other edge up to the cylinder line; the line should coincide with the edge of square. If it does not, the line should be shifted forward or aft until it does. The line will now give the true position for center line of cylinder. The alignment of cylinder can be verified by measuring from sides of stuffing boxes, or better, from lower counterbore.

If the cylinder itself is found out of line it will be necessary to adjust the cylinder feet and columns, or the whole cylinder may have to be moved. The cylinders are, of course, accurately aligned when the engines are erected, and should not change unless some part of columns or engine seating has been shifted or displaced.

The cylinder line being established, the crosshead guide

can be lined up. This must be exactly parallel to the line of cylinder. This can be found out by calipering or using a wooden stick to measure distance of surface of guide to center line of cylinder at several points along its length. Then the guide can be adjusted so that it is parallel. The surface of guide must also be perpendicular to a plane passed through the line of cylinder at right angles to the line of shaft. To determine this take a square and rest one side vertically on guide so that the edge of square just touches the cylinder line. Keep the corner of square in the same place and move square around. If the edge of square continues to touch the cylinder line the surface of guide is perpendicular; if not, the edge of square will move away from the line. The guide may then be moved till it is parallel. This being done, the standing parts may be considered as being in line.

Lining up Connecting Rod and Slipper. — The slipper and slide should be brought to a true surface, the surface plate being used for this purpose. The surface plate receives a thin coat of red lead; the slipper is placed thereon and moved back and forth. The slipper is then removed and the red spots scraped, and the process repeated till a good surface is established. When the surface is very bad it should first be trued up on a planer. Where no surface plate is available a straightedge can be used by placing it on the surface in various positions and noting where the high points of the surface exist.

The piston and rod being put in, the crosshead can be attached and the proper clearance between the guide and slipper obtained by liners. The alignment should be such that when the piston moves up and down centrally the slipper should just touch the guide through the whole stroke. The piston should then move centrally through the stuffing box.

Connecting Rod.—The connecting rod should swing in a direction at right angles to the line of shaft and should come fair over the crank-pin journal. To test this, observe whether the clearance between crank webs and end of crank-pin bearing is the same at each end. If the rod does not swing at

right angles to shaft the crosshead bearing may be at fault. The two bearings of crosshead may not be in line or one may be bent. If the bearings are all right the fault may lie in that the surface of slipper is not parallel to the axis of crosshead. This can be found out by putting slipper on surface plate and measuring distance to the center of journals. If they are not the same the slipper must be aligned by scraping or by fitting liners in proper place. If connecting rod does not come fair over the crank pin the fault may lie in a wrong adjustment of thrust collars, or the shaft couplings may not be properly brought together. Sometimes it may be necessary to shave off one end of crank-pin brass in order to get the lower end of connecting rod in.

Thrust Shaft.—This should be aligned at the same time with the shaft, for on the proper adjustment of the collars will depend the proper location of the whole shaft in a fore-and-aft direction.

There should be a uniform clearance for all collars. This clearance may be measured by inserting feeler gauges. The clearance varies from No. 24 to No. 29, American standard gauge. The clearance at thrust should not exceed the clearance between the crank webs and the crank-pin brasses, for if it does the thrust may be transmitted to the crank-pin brass.

The clearance on collars of horseshoe thrust is adjusted by moving the nuts securing the collars. In adjusting, great care must be taken that the surface of horseshoe is at right angles to the line of shaft. This can be done by making the clearance at each side exactly the same.

The horseshoes are generally babbitted and, as a rule, require re-babbitting in the course of every few years. Care should be taken that suitable oil grooves are cut on the horseshoes. A wavy line over bearing surface of collar makes a very desirable groove.

Various subsidiary parts of engine will require relining from time to time.

The valve gear is susceptible, in general, to the same treat-

ment as the main moving parts. The principal parts to reline are the guide and the link. The guide must be in such position that the valve stems work centrally in their stuffing boxes. Links generally wear on the go-ahead end, the gibs being loose here while tight when at the center of link. To rectify this the high portion must be scraped and the gibs adjusted accordingly.

The reversing shaft, pump link and indicator gear require relining and adjustment from time to time.

OILING GEAR, FRICTION AND LUBRICATION.

Friction is the resistance of the surface movement of bodies in contact. The substances may be solids, liquids or gases.

In engineering we have mostly to do with the friction of solids. Every metal surface, no matter how smooth apparently, has small irregularities in its surface, and when two surfaces are in contact these irregularities interlock and produce the resistance to motion, friction, which manifests itself as heat.

Friction increases with—

Increase of pressure;
Irregularity of surface;
Softness of surface;
Elasticity of surface.

Solid friction depends directly on pressure; when no lubricant is supplied it is independent of the surface, that is, the pressure being the same.

For slow velocities the friction does not change; for high velocities it changes, but mainly due to other circumstances, such as greater heating.

Similar substances in contact have more friction than two different substances.

Hence to lessen friction, obtain hardness, lessen pressure, use different substances in contact, use lubricants.

By introducing a lubricant, such as oil or grease, between the two surfaces they are kept apart. The irregularities are prevented from interlocking; solid friction is prevented, and instead we have the internal friction of the particles of oil on themselves and the friction of the solid on the oil.

Spongy metals, as brass and cast iron and various compositions known as white metal or anti-friction metal, can be made very smooth; hence they are very good for bearing surfaces. Cast iron is an exception to the rule that similar substances increase friction.

Hard cross-grained cast iron wearing on itself has less friction than any other combination of metals; especially is this so where the only lubricant is steam, as in slide valves and steam pistons.

The reason for this property of cast iron is that the surface, when rubbed lightly so as not to produce abrasion, will become very smooth and also very hard.

White metal and brass obtain their properties of lessening friction from the fact that they can develop a very smooth surface. Owing to the difficulty of fitting such places as bearings of fast-moving engines cast iron is not used, but brass and white metal instead. Of these two, white metal is the more easily fitted and it is used on nearly all bearings of any size.

There have been disputed claims as to whether brass or white metal has the less friction. White metal can be more easily fitted and adjusted and made smooth, and it is in general use in the best practice.

Rolling friction is very much less than sliding friction, hence the introduction of roller and ball bearings now largely used on special machines and on naval vessels in gun and turret mounts.

Ball and roller bearings are being introduced in thrust bearings for marine engines, and the use may extend. It is very likely that, if made very carefully, they will be a success on thrusts, but their wide use will probably be checked by the additional complication and difficulty in adjusting after wear.

The best lubricants for metals are—

Lard, olive, sperm, light mineral oil, ...	Ordinary machinery.
Mineral grease, tallow, .	For slow speeds, heavy pressure.

Sperm oil, heavy petroleum,	High speeds, high pressure.
Refined sperm and petroleum,	High speeds, light pressure.
Graphite, sulphur, soapstone, mineral greases, . .	Heavy pressure.
Heavy mineral oil, graphite, vaseline, . . .	Steam cylinders.

Mineral grease, tallow and graphite are the best of lubricants, but are more difficult to apply and, as they have little or no flow, can not be well used where there is any considerable speed on moving parts, owing to the slowness with which these lubricants distribute themselves over the surfaces.

Graphite is not affected by heat, and hence can be used where parts are under temperature, such as high-pressure steam cylinders and cylinders of gas engines.

The requisites of a lubricant are—

Efficiency to reduce friction;

Body, to support heavy pressure;

Fluidity, to avoid resistance and to rapidly cover the bearing surface;

Resistance to and power to absorb heat;

High flash and burning point, so that temperature may rise without danger of the lubricant burning.

Another requisite is the tendency to saponify and form a lather, which has an excellent lubricating value, especially for high speeds and heavy pressures.

From the above it will be seen that no one special lubricant is the best for all various purposes. Each special service should have its own particular oil or lubricant properly suited to its peculiar conditions.

Lubricants are tested: for purity, by chemical analysis; for viscosity, that is body; for gumming action; for flashing and burning point; and for general efficiency in reducing friction in a testing machine where friction is measured by a dynamometer.

Among the different methods of lubrication the following may be enumerated:

1. Oil bath, where the bearing is completely surrounded by

oil forced in under pressure. This is often used on large bearings on shore, and is used on the bearings of marine turbines.

Fast-running engines, such as dynamo and blower engines, are fitted with forced lubrication; that is the cranks, piston rods and main bearings being enclosed within an oiltight casing, through which oil under pressure is circulated with an oil pump. This system is economical in oil, since the oil is used over and over again. Objections to its use are the difficulties of preventing the oil being drawn into the cylinders and delivered to the condensers and boilers.

2. Self-oiling bearing, where some part of the journal distributes the oil from a reservoir. In this class are the centrifugal oilers for crossheads and crank-pins, and the loose-ring self-oiling bearings used on high-speed machinery such as turbine pumps (DeLaval).

Bearings fitted with such devices, however, usually occupy considerably more space than can well be allowed in the limited spaces allotted to naval machinery.

3. Pad lubrication, as in the boxes on car axles of railroads; also tallow cups on some forms of large main and thrust bearings.

This type is good for large, rather slow-moving journals where grease or tallow can be used.

4. Automatic sight-feed lubricators. Here the oil cup or box is fitted with a screw-down valve by which the amount of oil can be regulated by opening or closing. The lubricators are usually fitted so that the oil, as it drops, can be seen; and thus the amount fed can be ascertained. This system is very largely used, and answers very well for stationary work, but n marine work the vibrations and movement of the engines jar the valves so that the amount of oil is changed or even shut off. They thus require constant watching and need more attention than wick feeds would require. They are not a thorough success on naval engines, though the present practice is to have oil boxes fitted with both automatic sight feeds and the wick feeds. This is done so as to have the sight-feed

valve for use at full power, when the saving of oil is not a matter of such great consideration.

5. *Hand Lubrication.*—This is, of course, the most simple, but in the present complicated engines many bearings are inaccessible to hand oiling, and it would entail a very great amount of labor to oil even a majority of bearings by hand. Hand oiling is the most economical with skilled oilers, as they will not care to handle more oil than enough to keep the bearing from running warm, which is the result desired.

When running slow or at moderate speed a great many of the smaller and less important accessible bearings should be oiled by hand.

6. *Syphon Wick Lubrication.*—Here oil boxes are fitted at various parts of the engine well above the bearings to be lubricated. The boxes are supplied by piping from a reservoir tank placed above the engines. From the boxes oil pipes are led, without pockets, to the points to be lubricated. The oil is admitted to each pipe by a syphon wick, and drops into an open-top funnel mouth, where the drops of oil may be seen. The amount of oil can be regulated by making the wick larger or smaller or by changing the height that the oil is carried in the box.

The wick is weighted at the end resting in the oil box, and the distance that the end of the wick extends into the pipe is adjusted by a copper wire which is hooked over the top of the pipe and twisted around the end of the wick. The end of the wick should be about at the level of the bottom of the oil box to give a proper flow.

The great advantage of the wick feed is that it is easily kept at an almost constant rate, since the jar of the engine has no effect upon it. It also does not allow any dirt that may get into the oil box to be fed down the oil pipe.

A carefully looked after syphon wick system, supplemented by hand oiling at lower speeds, is the most satisfactory system at present devised for lubricating large marine engines.

With wick feed the oil in the boxes should not be allowed to go too low, as the height of oil in the box changes the rate

of feed. Wicks should be periodically lifted to keep them clean; if dirt accumulates about their ends they will stop feeding.

Where oil is fed directly to a bearing and not through a wick or long pipe it is of great advantage to add from 2 per cent. to 4 per cent. by weight of flake graphite. The use of graphite mixed with oil allows a lighter to be used, since the addition of graphite increases the body of the lubricant.

For slow-moving heavy bearings or gearing graphite grease is perhaps the best lubricant.

Oil flows more readily as the temperature rises. This point should be noted and the wicks adjusted for any material change of temperature of the engine room.

A slight rise in the temperature of the bearing reduces friction; hence the oilers term "working heat." This working heat must, however, be kept within proper limits and not allowed to become the excuse for allowing a bearing to become hot.

Specially Prepared Oils.—Lubricating oils are often specially prepared for the service desired by a compounding of different oils or the addition of other substances. The best results, as far as providing for a lubricant with least friction, is obtained by getting the oil to saponify (that is to form a lather). To do this some water is supplied to the oil in the bearing. This may be done by the water service; but the water service supplies salt water, which does not form as good a lather and is liable to leave salt on the bearing. If a little fresh water is supplied to the oil pipe by means of a syringe or funnel at occasional intervals a good lather can be obtained with high-grade lubricating oil. Adding a little soda to the water will help the formation of lather.

In such places as thrust bearings, which run in a bath of oil, sufficient water to make the proper mixture is added directly to the bearing by pouring it in with the oil.

In adding water care should be taken not to add too much at one time so as to wash the oil out of the bearing.

Storing Oil in Tanks.—In stowing oil in the ship's tanks

care should be taken (1) that tanks are properly cleaned, (2) that each tank has a strainer over its suction pipe to oil pump, (3) that different oils are not mixed, (4) that oil when run to tanks is passed through a strainer; cheese cloth or a similar substance will answer for the purpose.

Overhaul of Oil Gear.—This consists in repairing leaky oil pipes, setting up on loose unions and securing screws, etc.

Wicks are cleaned by being washed in soda water. They are made from lampwicking.

Oil pipes are cleaned or scalded out by introducing a little soda and boiling water, but care should be taken that none of the soda reaches the bearing.

All oiling gear should be carefully inspected just before getting under way.

The following are the latest specifications for the lubricating system of U. S. naval vessels.

There will be fitted in each engine room a steam pump of approved capacity, with hand attachment, for keeping the distributing oil tank supplied with oil. This pump will draw from the storage tanks and discharge into the distributing tank, an overflow pipe from the tank being led to the pump suction.

There will be in each engine room, located at as great a height as possible, a 10-gallon copper distributing tank. The tank will be well tinned on the inside and fitted with a glass gauge, air cock and pump connections. Each tank will be connected to all the oil boxes on its engines by ½-inch brass pipes, iron-pipe size, leading in such a way that oil will flow to each lubricator.

All working parts of the machinery will be fitted with approved and efficient lubricators. Generally speaking, there will be cast-iron oil boxes secured to fixed parts of the engine at convenient points and having connections for distributing the oil to the various bearing surfaces. Each connection will have a sight feed, an adjusting valve and a wick feed. The sight feed will have a well-protected glass tube. All oil cups with wick feeds will hold sufficient oil for four

hours' running. The distributing pipes will lead to wipers on the moving parts or tubes in the bearings and guides. The oil boxes will be polished on the outside, and will be plainly marked opposite each pipe with the name of the part to which it leads.

Unions will be fitted where necessary, so that the oil pipes may be quickly taken down and cleaned, and each pipe will be connected to the bearings by a union joint. Each main crank pin will be oiled by cups carried on the connecting rod, taking oil from wicks overhead; the oil to be carried to the crank pins by brass pipes secured to the connecting rods. These pipes will have union joints where connected to oil cups.

Each main crosshead journal will take oil from an overhead wick cup.

The crosshead pins will be flattened at the sides, a ring of metal being left at the ends to prevent the escape of oil. The oil will be introduced through a pipe passing through the axial holes and led to the surface of the pins through radial holes, terminating in the spaces formed by the flats on the pins.

In addition, oil will be introduced, through a pipe and oil box on cap bearing, to top of pin.

The crosshead guides will be oiled by pins connecting with holes leading to above the middle of each guide, as approved, and oil boxes will be fitted under each guide, with draggers, secured to the slippers, dipping into them.

The top of the slipper and the crosshead guides will have a concave bevel to prevent lubricating oil running over the back of the guide and slipper when slipper overruns guide.

The upper end of each eccentric rod will carry oil cups on each fork, long enough to prevent waste of oil. Each link-block pin will be oiled by two wiper oil cups, and each pair of gibs for links by one wiper cup supplied by pipes from the oil boxes.

Each eccentric strap will have a long oil cup fed by a drip pipe, so arranged that the eccentric will be lubricated in all positions, or else an approved telescopic oiling gear.

In general, all the oil for the external moving parts of each engine will be supplied, as far as possible, from oil boxes with separate valve, sight feed, wick tube and pipe, as directed or required, for each part to be oiled.

There will be one such oil box on each cylinder, from which pipes will lead to the following parts: Valve-gear connection, crosshead pins, crosshead guides, crank pins and valve-stem guides.

There will be provided, where directed, an approved number of oil boxes, fitted similar to those described above, so that they may be used either with valve adjustment or with wick tubes for lubricating the main crank-shaft bearings and the eccentrics. These oil boxes will be placed in accessible positions, and at least 5 feet above the center line on shafts, so that oil may be fed to the bearing under this head. Each main bearing will have two oil tubes, and the oil pipes will be so led and oil holes so drilled as to introduce the oil at both sides of each bearing near the middle of the length of the bearing, and as near the division between the cup and lower brass as is possible. These oil boxes will be supplied through ½-inch pipes from the 10-gallon distributing tank in each engine room.

Suitable small oil cups will be fitted for introducing oil into valve chests.

There will be fitted small oil cups as approved, for supplying oil to valve chests of pumps, blowers and other auxiliary engines.

Each blower engine will have a continuous automatic lubricator of approved pattern. All working parts will have oiling gear of approved design, such as will permit of oiling without slowing. All the oiling of each auxiliary engine will be done by one oil box where practicable. All fixed oil cups will have hinged covers, with stops to prevent being opened too far. Moving oil cups, where necessary, will have removable covers. The supply of oils to various parts is to be easily regulated. All oil cups and their fittings, except such as are cast on bearings, will be of finished cast brass, or sheet brass or copper, as may be directed, with all seams brazed.

All fixed bearings will have drip cups cast on where possible; otherwise the drip cups will be of cast brass, properly applied. All moving parts will have drip cups or pans cast on engine frames where directed; otherwise they will be substantially made of sheet brass or copper, with brazed seams. All drip cups will have drain pipes and cocks of at least $\frac{3}{8}$-inch diameter, which can be used while the machinery is in operation.

All auxiliary machinery will be fitted, as directed, with ample drip pans, with drains to bilge.

WATER SERVICE.

Engines are fitted with a system of water service by which cool sea water may be circulated through a bearing or guide, or sprayed on any part liable to heating.

Collars or thrust bearings, main bearings and crosshead guides are now usually cast hollow to allow for the circulation of water. On some recent large engines the crossheads are fitted with a telescopic water service to enable water to circulate through these.

In order to keep the large amount of water, which comes through the water service, out of the bilge and crank pits, some vessels have the water service arranged so that the water which circulates through the main bearings, guides and thrust bearings is led to the suction of the circulating pump. As thus arranged, the water service for these parts is a closed circuit.

There are usually a number of hose connections on the water service so that a hose can be attached and cooling water led to any desired place in engine room. These connections may also be used in washing down the engine room.

The repair work that the water service will require consists of grinding in its valves, tightening up on joints and unions and rebrazing broken joints.

The water service should be shut off about half an hour before coming to an anchor so as to leave only clean oil on the surface of the bearings when engines are stopped.

The following are the specifications for water service of recent U. S. naval vessels.

There will be in each engine room a 4-inch water-service pipe, with a 4-inch branch to the main engine and a 2-inch branch to the shaft alley, and smaller branches leading to different parts of the engine and shaft, as follows:

One 1¼-inch branch with swivel joint for each crank-shaft bearing;

Two 1¼-inch pipes to each crank pin;

Two 1-inch pipes to each crosshead;

One 1-inch pipe to each go-ahead crosshead guide;

One 1-inch pipe to each pair of eccentrics;

One 1½-inch pipe to each thrust bearing;

One 1-inch pipe to each line-shaft bearing;

One ¾-inch pipe to each hollow brass or its equivalent in crank-shaft bearings;

Two 1-inch pipes to each air-pump engine and to each circulating-pump engine.

Each of the above branches will have a separate valve and will terminate, as may be directed or approved, either in a pivoted nozzle, a detachable spray nozzle, a short length of hose, or a permanent connection to the part to be cooled.

The water-service pipe will be connected so as to take water from the casing of the main circulating pump. There will be a stop valve in the water-service pipe near the connection.

The water-service discharge pipes from crosshead guides, main bearings and thrust bearing will connect to the suction side of the main circulating pump, so that the water may pass overboard instead of into the bilges.

There will be a system of water-service pipes for auxiliary machinery in any part of the ship, as may be directed. These pipes will be of approved size, thickness and arrangement, and will connect with sea valves or pipes where directed.

CIRCULATING PUMP.

The general type of circulating pump used is the double-inlet centrifugal, directly connected by shafting to the pump

engine. The details differ for different vessels, and can be ascertained by an examination of the blue prints of pump drawings kept on board ship.

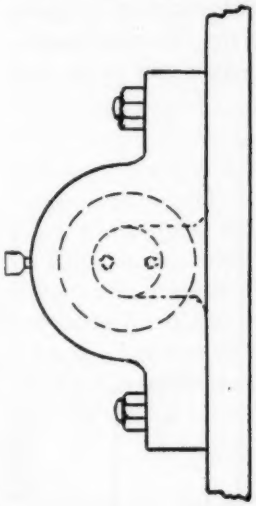


Fig. 17.

The casing is made in two parts, so that the upper part can be removed for purposes of examination.

On recent vessels the circulating pump also supplies the water-service pipe for engine room, and on some vessels the discharge from the closed portions of the water service, such as guides, main bearings and thrusts, discharges to the suction of the circulating pump. This latter arrangement greatly reduces the amount of water in bilges while under way.

There is often an end thrust on the shafting of the pump so that clearance of runner is destroyed on one side or the crank webs are liable to rub against main bearings. To remedy this, an end thrust bearing may be rigged by fitting a bracket to framing so that end of bracket comes abreast the end of shaft. In the end of bracket fit a set screw having a jam nut, the set screw to be in a position so as to be in line with center of shaft. When the set screw is adjusted so as to just touch the end of shaft it will prevent a longitudinal motion outward. (Figures 17 and 17A.)

The circulating pump requires examination from time to time; casing should be removed about once a year. The principal troubles met with are due to the corrosion of pump shaft and the runner getting loose on shaft. To prevent corrosion, zincs are fitted in the pump chamber and in the suction and discharge pipes.

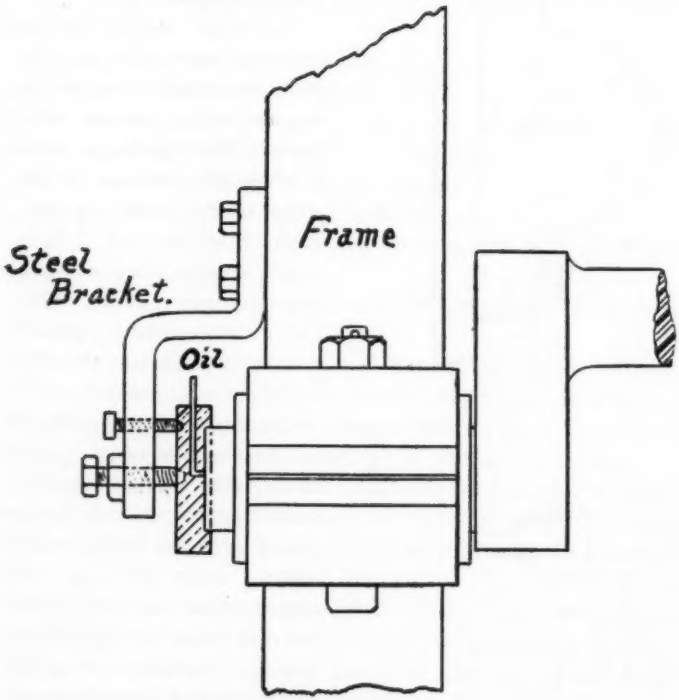


Fig. 17A.

Keys on runners must be properly secured by fitting them with set screws or other convenient way.

More or less dirt or marine growth may collect in the pump chamber; this should be cleaned out.

The lignum vitae bushing for bearings wear down in time and thus allow the shaft to drop out of line. They may be

brought back into place by turning bushing around, or better, by fitting a new bushing and placing shaft in line with that of the pump engine.

Circulating-Pump Engines.—These are usually compound, direct-acting engines. In general these engines require the same care as the main engine. The bearings must be adjusted when they become loose. The valve should be examined and kept properly set. The cylinders should frequently be opened and wiped out. They are usually fast-running engines and require fine adjustment. They run continuously for long periods without stopping, and, as a rule, very little difficulty is experienced with them. After engine has been overhauled it is advisable to run it for a short time to see that there is no danger of heating, otherwise it may be extremely unfortunate that when it is desired to get under way the circulating engine has a hot bearing which may delay the vessel.

CONDENSER.

The details can be seen by examining blue prints of condenser drawings kept on board ship.

The perforated baffle plates are provided to prevent the exhaust steam from striking the tubes directly and causing them to be bent out of shape or cut. The plates also serve to conduct the steam among the tubes so that all the surface may come into use.

The tubes are of composition, 70 per cent. copper, 1 per cent. tin, and 29 per cent. zinc. They are fitted with stuffing boxes and ferrule at each end, the glands being packed with corset lacing or cotton tape. The glands are provided with a shoulder which will prevent the tube from crawling.

Zincs are fitted in the salt-water side of a condenser to prevent corrosion from galvanic action, and should be fitted so as to secure a good metallic contact with the heads.

Condensers are provided with the following fittings :

Safety valves ;

Drain cocks for both salt and fresh-water ends ;

Air cocks to allow the accumulation of air to escape ;

A boiling-out connection to allow steam to be admitted to boil out the condenser;

A connection for admitting soda in solution;

A vacuum gauge.

The work of overhauling condenser consists of—

Keeping the cooling surface clean;

Preventing leaks from salt to fresh side;

Keeping shell tight so that there is no leakage of air from the outside;

Preventing the general corrosion and deterioration of parts.

The tubes in the course of time accumulate grease and dirt from the steam. To remove this deposit the condenser should be boiled out. This is done by admitting potash or soda into condenser through the soda cock, the soda being first dissolved in water. After the soda is in, live steam is turned into condenser through boiling-out connection. The mixture of soda and steam dissolves the grease, forming a soapy substance which can be drained off. Additional water is introduced to wash away the accumulation and remove the extra soda. The condition of tubes as to cleanliness should be ascertained frequently. The salt-water side of condenser in time becomes dirty from the accumulation of sand, ashes or gravel that has been drawn in and lodged in the condenser, so the salt side should be cleaned periodically.

Leaky tubes are caused by corrosion due to the action of salt water and of grease. Tubes sometimes appear to be cut due to the direct impingement of steam upon them. On this account baffle plates are fitted so that the steam does not strike the tubes directly.

In the course of time the action of the sea water causes the tubes to become brittle and liable to break. Tubes when made too light sometimes split, and pin holes may sometimes develop.

A most frequent form of leakage is at the glands at the tube ends. This is caused by the tube packing giving out or the gland not being set up tight.

All of the above leaks allow salt water to mix with the fresh, and their presence is indicated by salty water in the hotwell

or feed tank. Very often salt-water leaks occur through some form of salt connection on condenser that is not properly shut off. Many old vessels have a salt cock for the supply of extra feed. This should be disconnected to prevent danger of salt leakage.

To Locate a Leak in Condenser.—If in a hurry, take off hand-hole plates on condenser end, start up air pump and get a vacuum. Then take a lighted candle and hold it around the tube ends. Where the flame is drawn in there is a leak, and such tubes should have their glands set up on; or if it is the tube itself it should be plugged by screwing in a metal plug or driving in a wooden one. To locate the other end of tube at the other tube sheet, stick a long wire through the tube.

A more certain way of testing condenser is as follows: Close injection and discharge valves tightly. Take off end bonnets, or the hand-hole plates may be sufficient. Fill steam space with fresh water, usually done by means of hose through man hole. Then examine the tube sheets to see if there is any leakage through the tubes or around the joints, leakage being told by water trickling out. If a pressure can be put on condenser by means of a hand pump or by running up a stand pipe filled with water, the location of leaks is more certain. Care must, however, be taken not to fill L.P. cylinder with water in case no valve is fitted in the exhaust pipe.

The joints of the shell of condenser must be kept tight, as must also the joints in the exhaust piping. Shells of condensers are made in one course, but there are some that are made in several courses with joints. Leaks about joints may be located by holding a lighted candle about joint while air pump is in operation and noticing where the flame is drawn in, also by filling condenser with water and observing leakage. Leaks at joints may be repaired permanently by remaking the joint. This is, however, often inconvenient by reason of lack of time or on account of the great number of parts that have to be disconnected. Such leaks may be temporarily

stopped by caulking with cotton batting or by filling in opening with red-lead putty and fitting a clamp over joint to keep putty in place.

Leaks in the shell and joints are shown by poor vacuum. A very small leak may cause a very perceptible fall in vacuum. If the condenser is tight and the air pump in good condition a vacuum of 26 inches should be maintained even in hot weather.

The corrosion of tubes is guarded against by fitting zincs in the water chests. These zincs should be frequently examined and their proper connections made. The exterior of condensers should be kept clean and dry, and the rusting of the lower parts of lagging prevented.

Portions of water chests showing signs of corrosion may be preserved by coating with a light wash of cement. This is especially advisable around bolts or joints.

AIR PUMPS.

Air pumps are either attached or independent. With the independent air pumps the steam cylinders and reciprocating parts require the ordinary attention of such mechanical details. The detail of valve gear varies with the different styles of pumps.

The water end of air pumps is similar in most all cases and similar troubles are experienced. The principal points of overhaul are the following:

Cleaning and Repairing Valves.—The latest practice is to use a treble set of manganese-bronze discs kept in place by a guard and spring. In older pumps rubber valves are sometimes used. The valves and the seats become coated with grease from the oil carried over in the steam, and these must be cleaned frequently. Springs sometimes break and guards come loose.

The springs on foot valves should not be set up too tight, as the pressure necessary to lift the valves limits the possible vacuum. On the other hand, the spring must be tight enough to cause the valve to seat properly. The pressure on spring should not be over one pound per square inch on foot and

bucket valves, but the pressure on the head-valve springs may be more. The valves should have sufficient lift so that the circumferential opening is equal to the clear opening through the valve seat; as this is usually about 70 per cent. of opening the lift should be equal to .17 of the diameter, or roughly about $\frac{1}{4}$ the diameter of opening.

The buckets of air pumps are sometimes packed with bronze rings, but usually they only have grooves on the circumference. Plungers are usually packed with braided flax packing.

The presence of dirt, grit or pieces of metal in the pump chamber must be carefully guarded against and the formation of burrs or sharp edges prevented. Air-pump cylinders receive a natural lubrication from the oil which is always present to a more or less extent in the feed water.

The plunger rod must be kept tightly packed, and its packing frequently examined. Square flax packing is generally used here. Scoring or cutting of plunger rod must be carefully guarded against. Rods after considerable use may wear elliptical or become tapered. In these cases the gland cannot be kept tight properly, and it will be necessary to turn rod in a lathe till a true cylindrical surface is obtained.

DIFFICULTIES EXPERIENCED WITH PUMPS.*

The difficulties experienced with pumps are in most cases either in the water end or in the valve gear.

The following points will be taken up in connection with the water end.

Packing.—The proper operation of all pressure pumps depends in a very large measure on the packing of the plunger. For the packing of the plunger many different kinds of packing are used, but a thoroughly satisfactory packing for plungers seems hard to find, especially for high-pressure pumps.

Soft Packing.—The two principal kinds are square-braided flax and square cotton tucks hydraulic. Both these packings should be well treated with some lubricant to prevent excessive friction wearing the pump barrel.

The plunger must not be packed too tight to prevent ex-

*Some matters previously treated under the operation of pumps will be repeated here.

cessive friction. Cotton packing should be put on slightly loose, since it swells slightly when moistened. Cotton packing should be soaked in oil and coated with graphite before being put on.

Hard Vegetable Packing.—These are made from preparations of rubber and paper, and lignum vitæ rings are also used.

Lignum vitæ rings set out by springs in the packing now required by U. S. Navy machinery specifications for the low-service pumps, such as bilge and distiller circulating pumps. These rings give fairly good satisfaction.

Rings made of rubber or paper do not cause as much friction as the lignum vitæ.

Metal Packing.—Brass spring rings were used for feed pumps some years ago. They were not satisfactory, chiefly on account of the wear they brought upon the pump cylinders.

Packing rings of special white metal are sometimes used. These make a tight fit and do not wear the surface of the pump barrel as much as the brass rings do.

Outside-Packed Plunger Pumps.—In order to avoid packing troubles, outside-packed plunger pumps have lately been introduced for feed pumps. With these pumps the packing is put in a gland from the outside, and braided flax packing is used. The outside-packed plunger pump is considerably heavier than the ordinary pump, and if not well cared for there is danger of having considerable leakage at the glands. Special care must be taken to keep gland free from dirt and grit which would cause the plunger to become scored.

Wear on Pump Barrel.—When pump band has worn so that the surface is no longer cylindrical leakage will take place around the plunger. If this is badly worn, reboring is necessary. Pumps are usually supplied with a removable liner. To facilitate reboring portable boring bars are provided on some vessels, by means of which the pump barrel may be bored in place.

The stuffing boxes of both steam and water ends of pumps should be kept well packed. In pumps pumping fresh salt

water, such as auxiliary, circulating, flushing and distiller-circulating pumps, the water-piston rod should receive lubrication.

Valve Gear.—Considerable trouble is experienced with the valve gear. In general this is caused by neglect and bad treatment.

As a rule it is the secondary valve, which controls the operation of the main valve, that becomes stuck or fails to operate properly.

A few general rules as to the care of these parts may be given :

1. Remove valve gear frequently and clean off all parts and apply a little kerosene and graphite ; the graphite to lubricate the parts and the kerosene to prevent rusting.

2. Do not keep pump standing for any length of time without moving valve, as it may become rusted in place due to moisture which is generally present.

3. Do not keep drains open when not in operation. This allows access of air and will start rusting.

4. Do not supply cylinder oil to the valve while in operation, since this oil is liable to cake and cause the valve to stick.

5. Keep pump in continuous operation as much as possible. This applies especially to feed pumps, which if kept going continuously give little trouble, but do not work well when intermittently stopped and started.

The chief trouble experienced is caused by rings of rust or burrs being formed on the moving parts. The best way to remove these is to take off valve gear and clean it. The packing of the valve stem sometimes affects the operation of the valve. If the stem is too tightly packed the stem may not work readily and prevent proper action. If too loosely packed the valve stem may drop of its own weight before it should, and thus cause trouble.

The distance nuts on valve stem which regulate the length of stroke may be improperly set. These should be adjusted to give the proper length of stroke.

The cushioning valves may not be properly adjusted.

A detailed description of each style of valve gear will not here be given, owing to the great number of types met with. By application to the pump manufacturers a detailed description and rules for care, operation and adjustment can be obtained.

Valves for Pumps.—Feed pumps use bronze valves with springs and guards. These springs often break and the nuts holding the guards come loose. In overhauling the pump special care must be taken to see these parts properly secured by split pins and that the springs are set up to a proper tension.

The valve seats are in many cases removable and can be taken out and refaced when they become rough. The valves should always have a smooth bearing surface, and in overhauling the tightness of these valves should be tested.

Bilge pumps have rubber valves. In time these valves become soft and start to decompose, caused by the action of the oil. They may be turned and trimmed; but in most cases when they become soft it is better to fit new valves.

Location of Air Chamber.—The air chamber has a great effect upon the smooth working of the pump, but its efficiency depends very largely upon its proper location. The air chamber should be located so that the opening to air chamber is directly opposite to the direction of the flow of water. If it is at right angles to the direction of flow it is of very little use. Air chambers should be fitted to both suction and discharge sides of pump. Air chambers are specially needed on high-pressure pumps.

JOINTS AND PACKING.

Joints may be divided into two general classes: (1) Fixed joints like the joint between two flanges; (2) Sliding joints like those of a stuffing box.

For fixed joints a great variety of packings are used, the difference depending upon the pressure, the temperature and the nature of the contained fluid.

For high pressure strength is required, and hence packing for such purposes must be well strengthened.

For high temperature there must be material that will stand the continued action of heat.

For steam joints packing made of asbestos strengthened by being woven into cloth, or with an insertion of copper wire, is used. Asbestos is impervious to heat, but is softened by water; hence in steam joints where there is little likelihood of water being present asbestos sheet packing will answer well.

For steam joints where there is a likelihood of water a strengthened rubber packing should be used.

For water joints rubber packing should be used; for low-pressure ordinary sheet rubber will answer; for high-pressure cloth or wire insertion packing should be used.

Metal gaskets made of corrugated sheet copper are used for high-pressure flanges. The copper is pressed in tightly between the flanges and makes a metal-to-metal joint which, if it is properly made, will last long and remain tight. With this packing it is especially necessary that the flanges come square to each other and that the bolts are closely spaced and properly set upon all around.

Method of Making a Corrugated Copper Gasket.—The copper sheet is cut to approximate size of gasket; it is then corrugated, usually by being pressed between two heavy cast-iron formers. After being corrugated the gasket must be annealed, because the operation of corrugating it hardens the metal. The gasket is annealed by being slowly heated to a cherry-red heat and then cooled in water.

A thin coat of red lead should be placed on both sides of gasket, but it is wrong to put on a thick paste that will fill up all the corrugations, thus destroying the expansive properties of the gasket.

Wire gauze and red-lead putty, made of red lead, linseed oil and small pieces of hemp, is also used on large steam joints. The wire is flattened out and gives strength to the gasket, while the putty fills up all the vacant spaces between the wires.

Sheet lead is also used for making steam joints, but is is doubtful whether this is good for high temperatures.

A tight water joint can be made by simply using a coat of red lead and oil. Such a joint is not easily broken and has no elasticity and therefore may be caused to leak by vibrations.

Rubber has a tendency to grow hard and brittle under the action of heat; hence for high temperatures rubber packing is treated by special processes, and other material is added to avoid this tendency to break up. The special advantage of the rubber for packing purposes is its elasticity.

For steam joints the latest practice is to use copper corrugated gaskets for permanent work for high pressure. Strengthened rubber packing is used for small joints which have their flanges less carefully fitted and are more subject to vibrations. Strengthened asbestos packing is used for dry joints liable to be broken frequently, such as cylinder bonnets.

For high-pressure hot-water joints copper gaskets or wire gauze and red-lead putty are good; high-pressure rubber packing is also largely used.

For low-pressure steam a great many materials will answer, such as plain asbestos sheet, rubber with cloth insertion, or plain sheet rubber.

For high-pressure hydraulic joints leather is used, but this is not good for hot water. For hot water rubber insertion or copper gaskets are used. Low-pressure cold-water joints are made with plain rubber or rubber with cloth insertion, also canvas painted with red lead and linseed oil. The latter is largely used for hull fittings. Low-pressure hot-water joints are made with rubber, either plain or with cloth insertion.

For high-pressure pneumatic joints leather or paper is used.

Joints in Metal Fittings.—Bolt heads are made tight by putting a hemp grommet coated with red lead under them.

Threaded-pipe joints are made tight by coating the threads with red lead or white lead. Nuts are made tight on bolts in the same way. Red and white lead, however, cause the joint to freeze, so for threaded joints that may require removal graphite grease is used.

Tube joints in the headers of Niclausse boilers are made tight by using a special form of grease.

Where a permanent tight joint is desired preparations of metallic cement such as smooth on are of great value, especially for filling up cracks in metal or for tightening up on riveted or bolted joints.

Leaks in Joints.—If a joint is well made and the pipe kept under the same state of pressure all the time, it should last for many years; but joints that are intermittently hot and cold, moist and dry, will cause the packing to dry out, crack and rot. If the pipe is cold the metal contracts, and if in a dry place the packing dries out and shrinks. If it is a steam joint, when steam is turned on the joint will not be tight and the steam will work around the packing. If the packing is rotten or cracked portions of it may blow out. If the gasket is good it will swell by being moistened, and, after a time, will fill the joint tightly. Many joints that leak when water is present become tight as soon as the water is drawn off and the steam instead of the water comes in contact.

To Make a Joint.—The following conditions govern the making of a tight joint:

1. The flanges should come square to each other and be perfectly in line. The bolts must fit the bolt holes tightly and be exactly in line. If the flanges are not square to each other they will be open at some point of the circumference, and when the bolts are set up the flange will remain open or else be sprung; in either case the gasket will not be held tight and is in danger of being blown out.

2. There must be sufficient surface inside the bolt holes to enable a good bearing surface for the packing to be obtained.

3. The bolt holes must be sufficiently close so that packing is held securely all around. The greater the pressure the closer must the bolts be spaced.

4. The surface of the flange should be smoothly faced and, if not male and female, should be grooved.

5. For high-pressure steam joints male and female flanges should be used to prevent the gasket blowing out. (Not the latest practice.)

6. The gasket must be properly cut and fitted. The gasket

is cut by laying it over the flange, marking the holes, either with a scribe or by striking the packing over bolt hole with the ball of a hammer, and then cutting out the holes and shaping the gasket to fit the flange. Special knives and gasket cutters are often provided. Dies are supplied to cut the holes accurately. The holes should be cut slightly large so that the gasket can accommodate itself to the flange, and any bulging between bolts may be avoided. Care should be taken

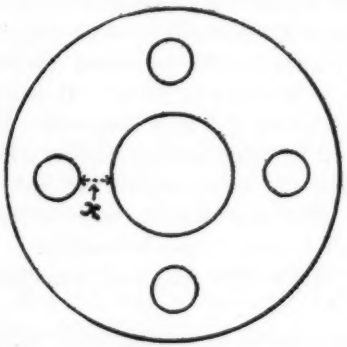


Fig. 18.

to see that the joint is not blanked off by omitting to cut out center hole. Precaution must be taken to see that the part of gasket in Fig. 18 abreast the bolt holes is not too thin or cut in any way. If there is not sufficient packing here the joint will leak at bolt.

Gaskets should always be coated with graphite and tallow on one side to facilitate the breaking of joint. The coating should be on the side of the removable part so that gasket will stick to the other. The joint may then be removed several times without renewing gasket. In order to hold the gasket to the other flange red lead or white lead is often used on the other side.

As a rule with high-pressure joints, the thiner the gasket the better the joint. However, where the flanges are not square to each other or well faced a thick gasket will be tight where a thin one would leak.

To fit a Gasket where there is strain on the bolts and where the cover or bonnet can not be removed.—This may happen in the case of a stop-valve bonnet supporting the yoke which holds the valve stem. Suppose valve must be kept closed and the joint packed.

The nuts are slacked back gradually, but the valve kept down on its seat by turning handle as nuts are slacked. Sufficient space being obtained, the old gasket is torn out and a new one cut, as shown in Fig. 19 (published in "Marine Engineering"), is worked around in place. The nuts are set up, the valve stem being turned to allow this, and then the joint

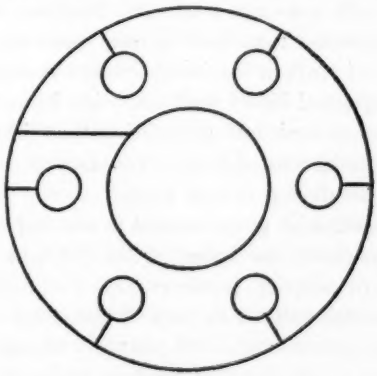


Fig. 19.

is packed. This method may also be used where to fit a gasket ordinarily would necessitate the removal of a great many parts, but where the flange might be backed off far enough to put the gasket in as above.

Joints Without Gaskets.—Such joints are ground in and require careful adjustment and a mechanical fit. Some examples of these are in cocks and valves, some tube caps on boilers, and tubes and headers on some boilers (Niclausse). The tightness of these joints will depend especially on the accuracy with which they are fitted.

COAL-TAR PAINT.

By Civil Engineer A. C. Cunningham, U. S. N., Member.

While on the New Orleans Naval Station the writer's attention was frequently attracted by the fact that when the Southern Pacific steamers entered the port of New Orleans their hulls, from the water line for a considerable distance up, were painted with common coal tar. Further inquiry developed the information that coal tar was quite commonly used on the bottom of craft in the New Orleans section.

There is a general belief that the acids in coal tar are injurious to iron or steel, and a tradition that they can be neutralized by the addition of lime. The fact that raw coal tar is practically non-drying is well known.

Coal tar is used with great success in coating cast-iron pipe, most generally under the name of the "Angus Smith Coating;" in modern practice, however, this coating is really coal tar, thinned occasionally with linseed oil, when continued use makes the bath too viscous. The success of coal tar on cast-iron pipe is due to the fact that the iron is dipped into a warm bath and is at a black heat; these conditions drive off all the volatile matter and the tar becomes a lacquer. Pipe so coated will appear as fresh as new after being buried in the soil for years.

While the common coal tar was under observation and inquiry a patented coal-tar paint was offered for sale on the station at fifty cents per gallon; the paint had the appearance of a high-grade asphaltum paint. Examination of the formula by which the paint was made showed that it consisted of coal tar with some twenty different substances added in various proportions, evidently in no scientific manner, but after a "cut and try" method. The substances constituted two groups, one of which was a dryer and the other an acid neutralizer.

With this information as a basis and the tradition that lime would neutralize coal tar, a series of experiments were made to develop a paint, using turpentine as a dryer. The experiments were a failure. It was found that the lime, whether quick or slaked, spoiled the mixture, and the result was a non-lustrous coating of no durability.

In the next series of experiments Portland cement was substituted for lime, the turpentine still being used; an improvement in results was secured, but the paint was not entirely satisfactory.

At this stage of the proceedings information was received that common kerosene oil would act as a dryer to coal tar, and another series of tests were started on this basis and the desired results were reached.

Since the writer left the New Orleans Naval Station the mixture has been applied to the bottom of the floating dry dock, and the next self-docking will give an idea of its value for under-water work.

The final mixture was tested by the Government chemist at the Washington Navy Yard, and pronounced non-injurious to iron and steel.

The tests of the mixture were carried still further by the writer while stationed at the Naval Academy, and a perfected method of mixing developed.

The proportions of the mixture are slightly variable, according to the original consistency of the tar, the use for which intended, and the climate in which used. In general it will vary between the following proportions, by volume:

	Coal tar.	Portland cement.	Kerosene oil.
New Orleans mixture.....	8	1	1
Annapolis mixture	16	4	3

The Portland cement should be first carefully stirred into the kerosene oil forming a creamy mixture; this mixture is then carefully stirred into the coal tar. For the best results it should be freshly mixed and kept well stirred.

With this tar paint old leaky tin roofs at the Naval Acad-

emy, on buildings that were to be torn down in the course of two or three years, were made watertight; on the worst places a coat was applied, then muslin was stretched over the wet paint, and, after drying, an additional coat of paint applied over the muslin. Temporary sheds that were covered with tin taken from demolished buildings had their roofs made watertight and sightly.

This coal-tar paint is the only mixture that the writer has ever found that will unquestionably stick to and thoroughly cover galvanized iron.

The cost of this paint will vary from ten to fifteen cents per gallon, and when it is considered that even a "fair" linseed-oil paint will cost from eighty cents to one dollar a gallon, the great economy is apparent.

Linseed-oil paints have no durability in salt water or in a very salt atmosphere, on account of the solvent action between the salt and linseed oil. The tar paint, however, being composed of purely mineral materials, is quite stable and much more durable under the same conditions.

Red-lead paint is an exception to the ordinary linseed-oil paints as regards durability. Ordinary oil paints dry by atmospheric oxidation from the outside in, assisted by the dryers or oxidizers that are added and whose ultimate effect is the shortening of the life of the paint. The red-lead paint is dried by the oxidizing action of the red lead itself, and the resulting coat resembles to some extent an amalgam, and is very hard and durable.

After a series of experiments on the New Orleans floating dry dock the writer found the following proportions to give the best results in red-lead paint.

Red lead, pounds	100
White zinc, pounds	20
Raw linseed oil, gallons	5
Turpentine Japan, pints	3¾

This should be used the same day that it is mixed, and if passed through a common hand paint mill after being mixed as well as possible by stirring, will be all the better.

For three-coat work on a clean, new surface the best results will be had by using 60 pounds of lead in the first coat, 80 pounds in the second, and 100 pounds in the last.

The white zinc prevents the paint from "crawling," and the raw oil is used on account of the intensely drying property of the red lead.

Boiled linseed oil is used in ordinary paints on account of its being already partially oxidized and thus drying more quickly.

The glaring red of red lead may be modified by the addition of a slight amount of lamp black without injury to the paint.

Ready-mixed red-lead paints are a snare and a delusion; true red-lead paint will harden in a few hours in a vacuum, and in consequence a ready-mixed red-lead paint has had the oxidizing property of the red lead destroyed and is reduced to the status of a common linseed-oil paint.

True, red-lead paint will cost from one and a half to two dollars a gallon, but if properly mixed from pure materials, carefully applied, and covered with a protecting coat of good common linseed-oil paint whenever it begins to show through will last for many years.

The purity of paint materials is a subject in itself, but as regards the coal-tar paint the materials are so inexpensive in themselves that no adulteration need be feared.

It goes without saying that a metallic surface that is to be painted should be as clean and free from scale as possible. New iron from a mill is generally painted too soon, as the mill scale is very tenacious for a long time, but will ultimately come off and bring the paint with it.

A rust *scale* is bad to paint over as mill scale. A light newly-formed rust is not injurious to paint over, provided there is not too much for the paint to reach the metal proper.

The theory that when any rusted surface is painted over the oxidation will continue on into the metal without extraneous assistance, is, on the face of it, erroneous, as iron oxide is a very stable combination. The conditions for rusting re-

quire the presence of both air and moisture; in dry air or pure water iron will not rust. Painting over a rust *scale* insures the contact of a porous layer containing both air and moisture against the pure metal, and rust scales continue to form under the most favorable conditions until the swelling of the oxide causes the whole to fall off. The black surface which then appears is sometimes mistaken for the mill surface of the metal.

The writer makes no unusual claims for coal-tar paint, but believes that it is better than most cheap paints of unknown origin and materials. It has been sufficiently demonstrated that it is of some value, its low cost is attractive, and in emergencies it might be of great use.

NOTES.

GASOLINE MOTOR BOATS.*

By M. M. Whitaker.

In considering the possibilities of the motor boat for naval purposes, it would probably be better to go over the ground already covered, and get some account of the work that has been done in this line, and also that projected.

In March, 1904, John E. Thornycroft, of London, read a paper before the Institution of Naval Architects which brought the possibilities of the internal-combustion motor officially before the society. His paper was wide in its scope, covering broadly the different points of internal-combustion motors which could be used in torpedo boats, with which his paper had to deal. He divided the possibilities into three: 1. Engines which drew their combustible mixture at atmospheric pressure; in other words, gasoline. 2. Engines which vaporize their mixture through heat or spray; in other words, kerosene and heavy-oil engines. 3. Engines furnished with gas from producers. In his paper he used for comparison a first-class torpedo boat and a boat of smaller dimensions and power but with internal-combustion motor. In diagrams accompanying the paper, he showed the motor torpedo boat driven by four six-cylinder gasoline motors actuating two propellers. The saving in space occupied was about 20 per cent; that is, the space devoted to machinery in the ordinary torpedo boat is about 50 per cent, while the gasoline engines with their reverse gear would take up about 30 per cent. Mr. Thornycroft's paper only went into the subject tentatively, and showed possibilities without advocating any specific motor or system. He seemed inclined, however, to favor either the gasoline engine or the producer-gas engine. His firm last year constructed a

* Paper read before annual meeting of the National Association of Engine and Boat Manufacturers.

40-foot open boat with an engine working on kerosene. This boat was called the *Dragon Fly,* and was fitted with torpedo-dropping gear, and attained a speed of about 18 knots per hour. The engine was placed forward, under a hood, with an open cockpit just abaft it. This was followed by the tank for fuel, and the torpedo was carried on the after deck. It would seem that boats of this type are too small for the service intended, and as they are open, are liable to be swamped when going in any kind of a sea.

The firm of Yarrow & Co. have just produced a 60-foot vedette boat equipped with triple screws, the two outside being driven by two four-cylinder, $6\frac{1}{2} \times 6$ motors, coupled direct to the shaft. The center screw is driven by one motor on similar dimensions, equipped with reverse gear. These motors are of the light high-speed variety, running at about 1,000 revolutions per minute, and the form of the boat is similar to that of *Napier II,* that is, very shallow draught, and having a truncated bow and flat bottom. The forward compartment is covered with a hood, and is intended to shelter the crew. The second compartment contains the motors and is also covered by a hood. The gasoline tank is located on the deck aft, the deck being dropped, and the tank fills out the form of the hull so that its presence would hardly be noticed. At the same time any leak would drain overboard. It is intended to mount a torpedo tube on the after deck, and as the boat complete will weigh only about thirteen tons, it will be possible to carry a number of them on the deck of a warship, to be launched, of course, at a suitable time. The boat has attained a speed of over 30 miles an hour, but the chief objections to her seem to be her shallow draught and flat bottom, which would make her a bad boat in a seaway, the opening of the engine room affording inadequate protection for the motors. The type also being of the high-speed variety would seem to be unsuitable.

Mr. S. F. Edge, selling agent of the Napier motors, a couple of years ago placed the *Napier Minor,* a 35-footer at the disposal of the Admiralty for experiments, and she was used

in the fleet maneuvers of that season in carrying messages between ships and the shore. For this service she seemed to be well suited, and it is to a certain extent on her success that the other experiments have been carried out.

The French Government has just issued conditions for a competition for naval service boats propelled by internal-combustion engines. They have laid down a series of rules governing the competition, and restricting the class of boat in many ways. The boats under the competition must be in the neighborhood of 60 feet long, decked over, which gives adequate protection for the motor, quarters for officers and crew, must carry anchors, gear and several small rapid-firing guns, with other ammunition. The speed must be at least 12 knots per hour, and the engines, of the kerosene type, are to be supplied by the French Government. They are of the heavy slow-speed variety, and under the conditions outlined very little room is left for the designer to exercise any discretion.

The Russian Government has probably more internal-combustion boats than any other, several, as is well known, having been built here in the United States, of the same type as *Gregory*, a 90-footer, equipped with two 300-H.P. motors. They also have several built by Belgian and French firms for light dispatch service, as for instance, one 40-footer fitted with Germaine motor, and several built in Germany.

Our own Government has not gone into the matter very deep up to the present, although they made experiments at one time with the *Standard*, and have since placed orders with a number of firms for motors of different classes. One branch of the naval service, however, in which the gasoline motor has already made much headway is in submarine boats, those of England and our own Government being all supplied with this form of power for use on the surface. I would refer to another type of motor which, while it has not been as yet in naval service, is interesting as showing the possibilities for lightness in motors. That is the motor of *Antoinette III*, an eight-cylinder, 5.9 × 5.9, made in V form, and giving on the

brake somewhere between 100 and 120 H.P., at a thousand revolutions. *Antoinette III* is equipped with two of these motors, having a total weight of only 750 pounds, and has shown a speed of over 30 miles an hour over long distances. This motor is remarkable for its light weight and throughout a season's hard racing on the continent has never failed to function properly, except on one occasion, at the last race of the season, held in the latter part of December. Here, it is said, it failed to go because proper provisions had not been made for cold weather.

This covers about all that has been done so far in fitting gasoline motors for the naval service, but I believe that the future will see their adoption in larger numbers and for more varied uses. Taking these uses in the order of their probable adoption, I should say that the first form in which it would be generally adopted would be for the ship's working boats. There are many motors on the market today suitable for this work, and they certainly can give no more trouble than the ordinary steam launch with which warships are generally equipped. From personal experience I know that these are constantly giving trouble and are a sore point with the officers of the engineer's staff. For boats of this class it will require a motor of medium speed and weight, but the motor must have adequate protection. This is the particular point where all the racers and high-speed boats have proved weak. In other words, the motor must be fitted in a compartment that is watertight. I do not think that a boat of the small high-speed type can ever be used as a torpedo boat. In the first place, in order to get the requisite speed in so small a hull, it is necessary that the weight of hull and machinery both should be cut to the limit in order to provide carrying capacity for the weight of a torpedo and its launching gear. This seems to have been entirely forgotten in the calculations of those who have dreamed of this class of boat, but aside from this, the light weight of the machinery and hull makes it unsuitable for the hard usage which it must undergo in the naval service.

The next class in order of probable adoption would be the vedette torpedo boat, ranging in length from 60 to 90 feet, and having motors of medium weight, medium speed, variety of from 300 to 600 or 700 H.P. In this class it is possible to arrange weights on a practical basis and make a boat which would be useful and in many respects more economical than a similar boat propelled by steam, while its cost would not be high, and a large fleet to operate from a base on shore or one or two carried by each vessel of a fleet would make a formidable array for any enemy to meet. The form, however, must be that of a torpedo boat, strong and light, and freaky features must be cut out. The probability of carrying internal-combustion motors up to the torpedo-boat class would seem to be possible only through the use of either heavy oil or alcohol motors or motors of the suction gas type, and for this reason: That the fuel consumption of the gasoline motor of large sizes becomes so large and expensive as to render its use improbable. Of course the Government can afford to do most anything it wants to, but, like the private individual, it does not want to go into an unnecessary expense. With the increased use of gasoline in multitudes of small motors, the price will undoubtedly advance to limits which will make its use almost impossible. Anyway, the naval service of this and other countries seems to have a prejudice against the fuel as dangerous. They seem to forget that they carry all the time explosives which are a thousand times more dangerous than gasoline could ever be, and experiences of the Government with it so far certainly do not justify their fear of it, as it is used in submarine boats, and the only instances of explosion have been through gross carelessness on the part of those handling it. If the handlers of the explosives had been as careless as those who had charge of the gasoline, the results would have been still worse.

For torpedo-boat service requiring powers well up in the thousands, it would seem that there are difficulties to be encountered with the use of gasoline as fuel, or with any internal-combustion motor in fact, that will require engineering of

high order to overcome Still, they are by no means insurmountable, and if the same thought is given to these problems that has been given to the development of steam, I believe that there is no doubt that they can be successfully overcome. The use of internal-combustion motors has so many advantages in the way of economy, saving of weight and hence increased speed, etc., that such difficulties as reversibility and ease of starting should not be considered. These can undoubtedly be overcome, in either case, by the use of compressed air. As I stated a little while ago, the saving of space is about 20 per cent, and the saving in weight even allowing that the propelling motors themselves would weight the same as a steam engine, is a big item, as the weight of the boiler is saved.

Now, as to some idea of what the successful motor will probably be. For large powers it will probably be a double-acting motor, having water-cooled pistons, and I believe on a two-cycle principle. The weight should be kept down to at least 25-30 pounds per horsepower, by the use of good engineering and suitable materials and might be even less. I should say that a piston speed of a thousand feet a minute would not be out of the way, and that the probable revolutions would not be over 600. In fact, this figure is one which should be closely adhered to except for the smallest class of tenders.

I regret that a greater time has not been given me for preparation on this subject and that I am only able to give an outline of what seems to me to be an important field of development for the gasoline motor and one in which it is bound to take a premier position within the next ten or fifteen years. —"The Nautical Gazette."

PROGRESS WITH THE MARINE TURBINE.

Almost every month that passes brings some addition to the literature in connection with the development of the steam turbine. The subject has been a most popular one, both for authors and those who join in discussions at the meetings of the various technical societies. It must be confessed that in

many cases there is a great deal of repetition. During the last few months we have had more than one text book dealing in a more or less similar fashion with the evolution of the various types of steam turbine. The greater part of these works are filled with data, and records, chiefly of interest to the designers of turbines used in connection with land plants. They are of interest and service also to those who may be placed in charge of such plants. Such works will probably become recognized authorities in due course, but for anything really new and stimulating we must rely upon the literature which issues from the various engineering societies. A paper recently presented to the Liverpool Engineering Society deals in a most interesting manner with the application of the Parsons turbine to marine propulsion. It is not, of course, the first paper of this sort that has been read, nor does it contain everything of an entirely original nature, but there are one or two facts brought to light which are well worth notice. First of all, we are glad to be reminded that the first compound steam turbine, built only twenty-five years ago, and developing a modest 10 H.P., after performing a great deal of useful work for electric lighting purposes, is now safely housed in the South Kensington museum. Then we are glad to observe, as the latest estimate, that the total horsepower of Parsons turbines delivered and on order is about 1,500,000, and that some of the units ordered are upwards of 10,000-H.P. It is also worthy of record that just as the success of the turbine on land was rendered possible by the unremitting devotion of Mr. Parsons to problems in connection with the very quick-revolution dynamo, so has the success of the marine turbine been brought about largely and the difficulties overcome by means of interesting, instructive and inspiring experiments and research conducted by Mr. Parsons with model screws. The story of the *Turbinia* and the ill-fated destroyers *Viper* and *Cobra*—the former having attained the almost unrealizable speed of 37 knots an hour—is now a matter of history, as are the later successes of the turbine steamers employed for passenger service by various railway companies

and the Allan Steamship Co. We have before us a paper delivered some time ago by Mr. Parsons before one of our technical societies in which he had the temerity to suggest that Atlantic liners would one day be fitted with turbines of 30,000 I.H.P. We well remember the reception which this statement received in the hands of the many incredulous engineers. It is not so very long ago, yet now the new express Cunarders under construction have more than double the horsepower mentioned.

It must not be forgotten that what we might also call an auxiliary invention has been instrumental in bringing about the triumph of the steam turbine. A good vacuum is absolutely essential for economical steam consumption. By means of the vacuum augmenter a rise of 1½ inches was obtainable in the *Manxman*, which represented nearly 8 per cent addition of economy. The steam used by the jet in the augmenter was about 1½ per cent, thus leaving a gain of over 6 per cent.

It is news to many of us that there are at the present time twenty-five leading ship-building and engineering firms in this country, as well as several foreign licensees, who have acquired the right to build Parsons turbines. The original company cannot cope with the demand.

There is one point with regard to the marine steam engine which will doubtless be settled at no distant date, and that is in connection with the economy to be gained either by wire-drawing or superheating the steam. It must be confessed that most marine men do not regard the superheater with any feelings of delight, and the practice, at any rate, in the Royal Navy, has been to generate steam at a comparatively high pressure and to let it be wire-drawn down so that it is in reality used as superheated steam at a lower pressure to that at which it is generated. We have seen no records of this being done with the marine turbine, and it is quite obvious that this method would entirely eliminate any danger due to very wet steam, or rather steam with water injuring the blades; it is a subject that has probably been well considered. We await with interest the results of official trials in connec-

tion with the marine turbine, because we feel that there is taking place a silent revolution in naval matters which will affect in no small degree a numerous class of engineers who go down to the sea in ships. It appears to be a far more simple charge to look after turbines than the huge reciprocating masses necessary to develop the same horsepower, and we fancy that we see here another example where there will be need for less highly-trained technical experts. The advantage, or disadvantage, of this state of affairs is, of course, quite an open question, but the marine engineer who does not read and note the signs of the times in connection with his work may one day have a rude awakening. The efficient engineer should certainly take the necessary steps to master the details and principles of the ubiquitous steam turbine. —"Engineering Times."

THE SPECIFIC HEAT OF SUPERHEATED STEAM.

An interesting discussion of recent investigations of superheated steam has been contributed to the "Rose Technic" by Prof. Frank C. Wagner, of the Rose Polytechnic Institute, in which he gives some new information concerning experiments previously reported in various publications. The information relates mainly to the experiments to determine the specific heat of superheated steam, beginning with those of Mr. J. H. Grindley made something over six years ago. His method was to allow saturated steam at a high pressure, from which the moisture had been removed by passing it through a separator, to expand to a lower pressure through an orifice. He took special care to prevent radiation losses by surrounding the chamber into which the steam expanded by a steam jacket filled with steam at the same temperature. A set of experiments was made in which all started with the same initial temperature but expanded into different lower pressures. Thus the total energy per pound of steam was always the same, since the steam started in the same initial condition and no external work was done other than that which is always

done in the generation of steam at constant pressure, and which is included in the values of the total heat given in the steam tables. Similar series of experiments were made, starting at different initial pressures. Prof. Wagner criticises these experiments because small amounts of moisture were apparently present, and because the thermo-couple for measuring the temperature of the steam was too close to the orifice.

A somewhat similar method was used by Griessmann. The steam, instead of being throttled by an orifice, was forced through a plug of silk fibers. The work thus done in overcoming friction was picked up again by the steam, so that the only external work done was that involved in the generation of the steam at constant pressure. This method is held by Prof. Wagner to be preferable to that of Grindley. Griessmann was also successful in removing the moisture from the saturated steam before expanding. Griessmann, like Grindley, regarded the variation in the specific heat as due to the temperature. A study of his experiments indicates to Prof. Wagner that the range of temperatures employed by him was too narrow to show with any certainty how the specific heat varied with the temperature. His experiments do show, however, a regular variation of the specific heat with the pressure. Prof. Wagner has averaged the results and converted them into English units in Table 1.

TABLE 1.—SPECIFIC HEAT AT CONSTANT PRESSURE, GRIESSMANN.

Pressure, absolute. Pounds per square inch.	Specific heat.	Average temperature. Degrees Fahrenheit.
14.22	.513	284.4
21.33	.524	288.4
28.44	.534	292.0
35.55	.541	295.6
42.66	.548	299.1
49.77	.561	305.0
56.88	.575	309.6
63.99	.585	314.4
71.10	.596	318.7
78.21	.600	323.6
85.32	.600	326.5

Experiments made by Prof. C. R. Jones in 1900 were performed by measuring the heat required to be removed in order

to cool a measured weight of superheated steam about 10 degrees centigrade at constant pressure. His results are given in Table 2.

TABLE 2.—SPECIFIC HEAT AT CONSTANT PRESSURE, JONES.

Pressure, absolute. Pounds.	Specific heat. Expansion.	Curve.	Difference.
14.51	.493	.484	+.009
34.51	.514	.514	.000
53.87	.541	.543	—.002
73.40	.568	.573	—.005
93.52	.608	.604	+.004
113.29	.632	.634	—.002

It will be noticed that both experimental results and those obtained from a "curve" are given. The curve is a straight line having the equation $c = 0.462 + 0.001525\ p$. The temperature of the entering steam averaged about 190 degrees centigrade and of the outgoing steam about 180 degrees centigrade, so that the temperature for which the specific heat was determined was substantially 185 degrees centigrade in all the experiments.

These results agree very fairly with those of Griessmann, showing a steady increase in the value of the specific heat as the pressure increases. The values at corresponding pressures are smaller than Griessmann's, which is in entire agreement with Lorenz's experiments about to be referred to. Lorenz found that the specific heat diminished as the temperature increased, the pressures being the same. The average temperature in Griessmann's experiment as given above was 305.2 degrees Fahrenheit, or 151.5 degrees centigrade, as compared with 185 degrees centigrade in Prof. Jones' experiments.

A direct method of determining the specific heat of superheated steam was used also by H. Lorenz. He superheated the steam by a gas furnace and then passed it through a water calorimeter at constant pressure. The steam used was condensed and weighed. The temperatures of the cooling water before and after passing through the calorimeter, as well as the quantity of cooling water, were determined. After making proper corrections for radiation losses the value of the

specific heat at the measured constant pressure is readily calculated. The values obtained are given in Table 3.

TABLE 3.—SPECIFIC HEAT, LORENZ.

Absolute pressure. Pounds.	Average temperature. Degrees Fahrenheit.	Specific heat.
27.5	405.6	.531
26.9	561.2	.492
61.3	401.7	.583
57.6	563.0	.484
56.0	642.0	.478
97.0	405.6	.622
97.0	534.7	.568
98.8	609.8	.555
127.6	432.2	.698
127.7	503.6	.640
127.8	596.8	.570

In these experiments both the temperature range and the pressure range were considerable, and there is no room for doubt that the specific heat varies with both the pressure and the temperature. It increases with increasing pressure and diminishes with increasing temperature.

The problem has been attacked from a still different standpoint by R. Linde. Making use of a new and careful determination of the specific volume of superheated steam at various pressures and temperatures, Linde has constructed an equation for superheated steam, and by the general well-known equation for specific heat at constant pressure he calculates the values of the variable part of the specific heat. Assuming that the constant term in the specific heat is such as to give Regnault's value of .480 at atmospheric pressure he calculated the specific heat at constant pressure to be as given in Table 4.

TABLE 4.—SPECIFIC HEAT, LINDE.

Pressure. Pounds.	Temperature, Degrees Fahrenheit.	Specific heat.
14.2	288	.483
28.4	292	.500
42.7	299	.515
56.9	310	.528
71.1	319	.541
85.3	328	.556

This also shows that the specific heat of superheated steam

increases with the pressure and the form of the equation used by Linde shows that the specific heat also decreases with the temperature when the pressure is constant. The equation proposed by Linde to represent the relation between the pressure, volume and temperature of superheated steam is:

$$p v = B T - p(1 + a p)\left[C\left(\frac{373}{T}\right)^3 - D \right]$$

where $B = 47.1$; $a = .000002$; $C = .0310$; $D = .0052$. By introducing the equation for the adiabatic of superheated steam he deduces for the value of the specific heat of steam at constant pressure:

$$c_p = A_n \left[B + 3p(1 + a p) \frac{C}{T}\left(\frac{373}{T}\right)^3 \right]$$

A is the reciprocal of the mechanical equivalent of a calorie in kilogrammeters; $n = 4.2$; p is the pressure in kilograms per square centimeter; and T is the absolute temperature in degrees centigrade.

Using this equation of Linde Prof. Wagner calculated the values of c_p for a temperature of 392 degrees and also for a constant temperature of 572 degrees Fahrenheit, with the results given in Table 5.

TABLE 5.—VALUES OF c FOR TEMPERATURES OF 392 AND 572 DEGREES FAHRENHEIT.

Absolute pressure, pounds.	Degrees.	Degrees.
14.22	0.477	0.472
28.44	0.487	0.476
42.66	0.497	0.481
56.88	0.508	0.486
71.10	0.520	0.491
85.32	0.531	0.496
99.54	0.543	0.501
113.76	0.556	0.507

A few months ago Prof. O. Knoblauch published a note concerning experiments to determine the specific heat of superheated steam, which were made in a quite different manner from any previously employed. The steam under pressure from the boiler was first passed through a series of chambers containing electrical heating coils where it could be super-

heated to any desired amount. A portion of this superheated steam was then passed through a copper coil immersed in an oil bath and still further superheated, after which it was condensed and weighed. The temperatures of the superheated steam, both before and after passing through the copper coil and the amount of electrical energy absorbed by the steam were determined and from these the specific heat was calculated. The energy absorbed by the steam in the copper coil was determined by finding the difference between the electrical energy required when the steam was passing and that required to maintain the oil bath at the same temperature when the steam was shut off. The numerical results have not been given, but it was reported that the tests were made at pressures of 2, 4, 6 and 8 atmospheres and at intervals of 50 degrees up to 350 degrees centigrade. The values thus obtained are stated to agree within 3 per cent with those calculated by the Linde formula.

Prof. Wagner has plotted the values of c in terms of the pressure for each set of experiments where the temperature was substantially constant. The results obtained by Griessmann and Jones agree very satisfactorily, and the results obtained by Lorenz show substantial agreement with the others, but are more erratic. The values calculated by Linde's equation agree fairly well with the curves given at low pressure in some instances, but at higher pressures the calculated values are smaller than those obtained experimentally. As a result of his study, Prof. Wagner is convinced that further experiments are needed, particularly at high pressures and temperatures, to settle definitely the values of the specific heat of superheated steam under the conditions now frequently met in commercial practice.—"The Engineering Record."

THE DYNAMICS OF SCREW PROPELLERS.

By Professor Robert H. Smith.

The subject of screw propellers seems to provide perennial entertainment to engineers outside the classes who have per-

force to design, to make, and to use them. To these latter classes they are continuously a cause of perplexity, astonishment, vexation and disappointment.

The only excuse offered for the present incursion into this vortex of confused discussion is that the intention is to conclude with a suggestion of actual constructional improvement. Whether this suggestion will be of practical value it will be for the naval architects and marine engineers to decide. It appears, at any rate, that the suggestion is new, or, if not novel, must be a revival of a proposal that has not hitherto had a fair hearing and has never been fairly put to the test.

Some amateur students of hydrodynamics err very greatly in forgetting that the equation $p + \frac{1}{2} \rho y^2 =$ constant does not represent any true law outside the limits of a steady stream flowing in a channel with fixed walls; that is, with walls which can neither do any work nor have any work done on them. If the walls do work on the flowing fluid that work has to be added—per proper unit of volume and of time or length of channel or other unit considered in the progress of the action —to the constant on the right-hand side. As it stands unmodified, the equation is the hydrodynamic law of a conservative system, that is, of a space whose total energy is neither increasing or decreasing by integral influx or outflow of energy. The lamentable way in which automatic engineers forget the real physical meanings of their formulas is well illustrated by the fact that the sole popular engineering application of this formula is to the case of flow through frictional channels in which energy is being steadily developed by loss of gravitation potential, and at the same time steadily spent in doing work on the friction of the walls. For this case the formula is, of course, modified so as to give correct results; yet the letters published in "The Engineer" show that many of the writers would like to apply it to working hydrodynamic machines without any modification at all. "Hippocrate dira ce qu'il plaira—mais le cocher est mort;" doctors may flourish equations which they do not understand as much as they please

—but the trust law of conservation will stop the machine if it goes on working without being fed with new energy.

Mr. J. Batey, infers from Rankine's declaration of 1865 that "the best propeller is one that drives the least water astern at the lowest velocity," that the "perfect screw drives no water astern and therefore with no velocity." Passing over the embarrassing substitution of "least" for "greatest," which is probably accidental on Mr. Batey's part, could any inference be more astonishing? The ship's resistance is prescribed and it equals $M\ S$ so that $M\ S$ can only vary inversely to each other; if S be diminished, M must be proportionately increased. But the kinetic energy involved in this momentum is

$$\tfrac{1}{2}\, M S^2 = \frac{(M S^2)}{2\, M} = \frac{\text{Prescribed constant}}{M};$$

and the only way to reduce this loss is to increase M.

Here it is interesting to note that Rankine's paper in "The Engineer" of November, 1866, as quoted in this issue of August 4 of this year, just doubles this kinetic loss. The error is probably due to the fact that people had then hardly got out of the old fashion of measuring the energy of motion by vis viva, or MS^2, instead of by $\frac{1}{2}\ MS^2$, which is the true kinetic energy.

It is worth putting down the calculation for the sake of a very simple graphic construction for the efficiency which the writer has not seen given elsewhere.

Equation I.—

R = prescribed ship resistance = necessary direct propelling force = repelled mass per second × new backward velocity of this mass = $M\ S = p\ A\ (V+S)\ S$.

where p = mass per unit volume of water, A = cross-sectional area through which the repelled steam passes, and V = speed of ship, and M the mass repelled per second with the newly impressed sternward velocity S.

It should here be carefully noted that there is involved in this the assumption that the water which is taken through the propelling machine has zero fore or aft velocity before

its entrance into the machine. This does not at all mean that it retains such zero velocity until it reaches and actually touches the propeller blades, or comes under their direct impulsive influence.

It means that the ship is steaming through quiet water; or that, if its progress is against, or along with, a current, everything is measured relatively to that current—both momentum and velocity of kinetic energy. V is the speed of the ship through the current, measured relatively to the water of the current and not relatively to the earth. But the new velocity S may be acquired long before the water repelled by the propeller blades actually reaches these blades. To illustrate from the analogy of a fan or of a centrifugal pump, suppose that the "suction" and "delivery" pipes are of the same area; then evidently there is the same velocity of passage through these two pipes. [In the case of the fan there is a small rise of pressure, and therefore of density, and a corresponding small decrease of velocity in passing from suction to equal-sectioned delivery, but the change is so small as not to affect the question materially.] The velocity of delivery has been wholly acquired before arrival at the blades of the pump. It is acquired through the down grade of pressure from the atmospheric pressure of the reservoir (plus or minus any water head in the suction pipe) to the partial vacuum pressure maintained behind the fan or pump blades by the dynamic action of these blades. The work of acceleration up to this intake velocity is directly done by the atmosphere which pushes the air or water into this vacuum; but it is eventually done by the fan or pump which has to do the same amount of work upon the atmosphere at the other end of the circuit, *i. e.*, at the far end of the delivery. This is the scientific answer to the inquiry as to "Elusion of Instantaneous Acceleration" made in the leading article in "The Engineer," of March 24. As pointed out in that article, no instantaneous integral acceleration occurs in any of these machines, fans, pumps or propellers, nor indeed in any place in nature. So far as principle and general outline of action are concerned,

there is no mystery whatever about the matter. The same accelerative work is done in the same way in the suction or supply part of every pumping machine, whether of the piston, rotative, induced jet, or any other type. There may be in many cases considerable obscurity as to the exact shape and varying sections of the stream lines along which this acceleration in the supply occurs. For instance, what practical or mathematical engineer has ever been able to work out the stream lines of the passage past the multitudinous forms of supply and delivery valves in use or through the ports and bends whose shapes vary only in the degree of their utter badness from the point of view of hydrodynamic efficiency? All that we do know certainly is that the lines of these valves and ports are commonly designed without any reference to the shapes, good or bad, of the fluid stream lines that they compel. Until the actual stream lines throughout the length of the intake can be definitely determined, it is impossible to see how exactly the acceleration up to the final inlet velocity is distributed. The precise mode of action from point to point remains obscure. But if by measurement we discover the actual pressure at entrance just before reaching the propelling blades, then with a speculative allowance for frictional and viscosity losses, we can calculate this entrance velocity.

Be it observed, however, that this entrance velocity produced by the lost head down to the entrance pressure is not necessarily in the direction of the useful working effort of the propelling blades. Practically it can never be so in any form of rotating pump, any more than it is so in a turbine. In the case of a ship propeller, it is certainly not axial, and its amount, therefore, is no direct indication of that of the direct axial discharge velocity S. Clearly, its magnitude may be many times that of S.

For a prescribed ship resistance R and a specified cross-axial section A of the repelled stream, Equation I is a quadratic for the calculation of the discharge speed S. There are two mathematical solutions, but one only has application to the ship problem: it is

Equation II.—

$$S = \frac{V}{2}\sqrt{1 + \frac{4R}{\rho A V^2}} - 1 \Big\}$$

from which the mass repelled per second may, if desired, be calculated as $M = \frac{R}{S}$.

From this Equation II it is apparent that so long as the ship resistance R remains in the same proportion to V^2, the axial discharge velocity S varies in constant proportion to V. the speed of the ship. If R increases more slowly than V^2, S increases more slowly than V; and if R increases more rapidly than V^2, the required S increases more rapidly than the speed. If throughout any range of speed R remains constant, as is nearly the case for a considerable range beyond the "critical speed" in Mr. Yarrow's recent interesting trials, then S diminishes with increase of V throughout that range. [This is easily proved by taking the differential coefficient of S with respect to V.]

The rate at which work is done usefully in propelling the ship is $R\ V = M\ S\ V$, and the rate at which kinetic energy is generated in repelling the mass M per second backward is $\frac{1}{2}\ M\ S^2$, calculating only from the necessary axial component of the "delivery" velocity. The ratio $\frac{1}{2}\ M\ S^2 \div S\ V = \frac{1}{2}\ S \div V$ is the minimum ideal inefficiency inseparable from the method of driving ships forward by thrusting backwards on the water. It yields the following measure of the

Equation III.—

$$\text{Ideal maximum efficiency} = \frac{4}{3 + \sqrt{1 + \frac{4R}{\rho A V^2}}}.$$

This may be represented graphically by a very simple diagram. Having a common center, two opposite circular quadrants are drawn with radii equal to 1 and 2. Along a tangent to the smaller quadrant is plotted from the touching point $\frac{2}{V}\sqrt{\frac{R}{\rho A}}$.

From the point s, obtained by this plotting, a line is drawn through the center cutting the two arcs in e and E. Then the efficiency of Equation III is $\frac{E_e}{E_s}$. The height plotted to s is proportional to $\sqrt{\frac{R}{V_2}}$, and so long as R varies at V^2 this efficiency remains the same.

The plotted height is inversely proportional to $\sqrt{A}$, and the diagram exhibits very graphically in what manner this efficiency is increased by increasing A, or the cross-axial section of the repelled stream of water. For any one shape of the section, such as the circular, $\sqrt{A}$ is, of course, proportional to its diameter or other lineal cross dimension.

It is not possible to assert that this calculation has any extremely near practical importance. The whole of the discussion in "The Engineer" this year has been devoted to the question—how does the propeller drive the ship? But the really difficult, mysterious, and extremely important question is how does the propeller waste energy without driving the ship? Calculations made by the writer show that in many cases more than 50 per cent of the actually indicated engine power cannot be accounted for by the sum of engine and thrust-block friction plus skin and wave resistance of the ship. The late Mr. Froude said that no more than 40 per cent of the power was commonly spent in usefully overcoming the ship resistance.

The question of what becomes of the rest is of immense importance, because if it could be saved, a saving of at least 30 per cent of fuel and coal-bunker capacity would be possible. Only a small proportion of this loss is accounted for by the above $\frac{1}{2} M S^2$. If Rankine's slip of the pen in putting this down as $M S^2$ has had any real effect in making marine engineers imagine that their propelling machines have any efficiency worth speaking about, that would be greatly to be regretted.

"Superintendent Engineer" speaks very positively about the actual existence of what he implies is a very large value

of S. Although cavitation, or partial vacuum, on the forward surfaces of the propeller blades almost certainly exists in greater or less degree under all circumstances, this does not in the least supersede the necessity of the $M\ S$ as an abutment thrust. The simplest way of viewing that point seems to be to remember that this cavitation vacuum exists between the propeller and the stern hulk of the vessel, and whatever advantage might apparently be derived from the lack of sternward pressure on the front surfaces of the propeller blades is lost by the equal lack of forward pressure on the stern bulk of the ship.

It is possible that this vacuum may die away even in the narrow space between the two in an upgrade of pressure towards the hulk; from the probable curvature of the water stream lines here some upgrade of pressure, although a small one, very likely exists here. But this would hardly affect the matter, because the water close to the hulk would in that case be practically dead water to be carried and driven along with the ship as if it were itself part of the hulk. The actual magnitude of S is not, however, directly proved to be great. "Superintendent Engineer" asks us to observe the immense back flow of water easily visible when a ship is moored fast and its paddles or propellers rotating; but this observation is quite irrelevant. The present writer has spent a great many hours in watching the progress of the flotsam both inside and outside the apparent limits of the wake-stream, from the propeller in ocean and Channel steamers, and, although there was usually clear evidence on the surface of a real backward S, yet this was not always so, and the apparent magnitude of S was never large. Of course, the relative speed at which such flotsam recedes from the vessel is great, but that is $(V + S)$. It is also to be remembered that one can only observe the surface water, and it is probable that the backward under-current is much greater than on the surface.

It is clear that the action of the propeller must to a very considerable extent modify the stream lines around the ship, and thus influence the wave resistance, and to a smaller de-

gree the skin resistance of the ship. It is conceivable that under certain conditions it might lessen this resistance; but the normal result of the propeller disturbance seems probably an increase of it. Is it not regretable that the intake of water from round about the stern hulk into the propeller should be left absolutely without any guidance of any kind? Such lack of guidance at entrance into the working machine must inevitably result in unnecessary eddy making, and also in the arrival of the water at the blade edges in directions deviating from the most economic proportions. What may be termed the inside surface of the intake channel is the stern of the hulk; but its lines are designed mostly without reference to true guidance into the working machine: they are modeled on the lines found to give least hulk resistance when the ship is towed at the bow and is unhampered by obstructions such as a propeller at the stern. The outside of this intake channel has no prescribed boundary. In no other hydraulic machine in existence is the inlet channel left so completely to form itself at its own sweet will, or in obedience to "natural laws" in contradistinction to artificial restraint. The essence of all good machines is to artificially restrain the motions of its parts along most economic lines, and this fundamental law of machine construction is completely neglected in ship propulsion by screw propellers.

What are the best guiding lines up to the front edges of a screw propeller is a difficult problem of design far beyond the scope of this article. It is one which the writer hopes may engage the attention of marine experts in future years, but which will not be settled by a single theoretical investigation nor by a single experimental research.

The delivery side of the machine is equally without guidance. It would be hard to say if this results in greater loss than the want of inlet guidance. What is certain is that the loss must be immense. The most elementary mechanical consideration of the thus unguided propulsive action of the screw blades shows that the velocity of delivery must have a large component that is not axial; and the first half hour of observa-

tion over the stern of an ocean steamer shows anyone, with eyes in his head, that this is the actual result. The whole of this transverse component of discharge velocity represents pure unnecessary waste of kinetic energy. If the angle which the discharge velocity makes with the axis be called δ, then the kinetic energy thrown away in the wake-race is increased from the necessary $\frac{1}{2} M S^2$ in the ratio $(1 + \tan^2 \delta.)$. If δ be 45 degrees, it is doubled; if δ be 60 degrees, it is quadrupled; if δ be 75 degrees, it is multiplied 14-fold. If S, T and C be the axial, tangential and centrifugal components of this discharge velocity, then the kinetic energy abandoned in this discharge is greater than that essentially necessary in the ratio—

$$\left(1 + \frac{T^2 + C^2}{S^2}\right),$$

whereas the useful propulsive thrust remains proportional to S only.

Suppose that it were found to be desirable for the mechanical efficiency of the machine as a mere driver of water that the discharge at the delivery tips of the blades should have these components in the proportions S, T and C, then if this discharge were received into properly designed fixed guide channels which deflected the whole velocity $\sqrt{S^2 \times T^2 \times C^2}$ into the backward axial direction, then the useful propulsive effort would be proportional to this whole velocity, instead of simply to S alone. That is, you can reduce your expenditure of kinetic energy in the wake to the scientifically essential proportion, and yet discharge your water from the delivery tips of the blades at any radial and tangential angles you may find desirable for the good working of the machine, if you will only discharge into a fair-lined exit channel deflecting the whole into parallelism with the path of the ship's motion.

So far as the writer of this article has studied the problem, it would not be possible, with screws as commonly designed, to get the desired propulsive effect without the blades imposing upon the water a tangential velocity. It may not be wholly imposed on it by direct contact during their passage; it may be partially, or even under possible conditions wholly,

imposed indirectly during approach to the blades along the supply stream lines. But it would be silly to suppose that the water at any instant in contact with any part of the blade surfaces, front or back, had any relative velocity normal to the blade. The whole relative velocity of the water actually touching must be parallel to the blade surface, and that of water passing near the surface must be more or less approximately parallel to it. The "absolute" motion of the blade is at every point along a helical path with velocity prescribed by V and the revolutions per minute, and the absolute velocity of the water equals this compounded with a velocity parallel to the blade. This relative velocity may under certain critical conditions be zero on either surface, back or front; that is, there may be masses of dead water carried round adhering, so to speak, to either back or front of the blade. Under other critical conditions of speed the tangential component of relative water velocity may just counterbalance the tangential component of absolute blade velocity. But away from such special and critical conditions, there must be tangential velocity, along with which must be generated centrifugal acceleration. This sets up outward radial currents, or radial components of velocity. The general direction of relative flow over the sternward faces of the blades is from near the boss at the forward edges towards the tips at the back edges. The chief supply streams must enter along the inner or more central parts of the cutting edges, little, or perhaps no, supply entering near the outer tips. Thus along the approach stream lines there is inward radial component of motion, and along the back surface of the blades outward radial components. Beyond certain critical rotative speeds co-ordinate with the corresponding traveling speed, these tangential and radial components of velocity must increase rapidly with the speed; and the critical speeds are probably low in all common forms of screw propellers.

It is in the abysmal depths of the vortices so arising that, according to the belief of the writer, are swallowed up the enormous losses of energy that are known to occur, but which

have never been measured or calculated. The vortices are probably quite unavoidable, but their kinetic energy may be recovered and utilized by improved design.

It may be utilized in either of two ways:—(1). By deviation, as already suggested, of the flow from the direction in which it leaves the blade edges, into parallelism with the axis; and (2), along with (1), by diminution of the magnitude of the velocity after it has left the blades by discharge through a channel of tapering section. This second method leaves not only the direction but also the velocity-magnitude of the discharge from the blades to the free choice of the designer of the propeller and of the engine. As in the evase discharge from a high-lift centrifugal pump, the high velocity and kinetic energy of discharge from the moving machine is converted into pressure energy or "head." The $M\ S$ as it leaves the propeller blades is insufficient to provide the desired R, or propulsive effort, needed for the speed of the ship; but the forward pressure on the walls of the fixed evase discharge tube due to the extra pressure created by the widening section and the consequent diminution of velocity and of kinetic energy, makes up the deficiency in R, and gives the prescribed R with less kinetic expenditure in the final $\frac{1}{2} M S^2$.

We have now arrived at the idea of a propeller with guided inlet channel and guided exit channel for the water. The guidance will be the more complete by the insertion of guide blades between the walls of the guiding channels. The whole machine has become an enclosed one, and its various parts, fixed guides and working blades, are capable of as strictly scientific design as are those of a high-class turbine, or of a Reynolds-Mather or a Sulzer high-lift centrifugal pump. The water is everywhere artificially constrained to follow the paths of mechanical efficiency; it is no longer violently tossed and swirled at the mercy of the wild tyranny of the free ocean. In a word, the propeller becomes a civilized machine, and drops its character of a barbaric warrior fighting rudely and blindly with Nature's stormy forces.

But, finally, now that the propeller machine is enclosed in a casing rigid with the ship's hull, there is evidently no reason why it should remain outside the fair lines of the skin of that hull designed to minimize ship resistance in accordance with the laws established gradually by nearly a century of practical trials and theoretical and experimental research. Outside these fair lines it is certainly a great disturber of the economic stream lines, to procure which is the sole object of the design of these lines. If withdrawn inside the hull its intake and its discharge must still be reckoned with as greatly influencing these stream lines; but its body, at least, is no longer an obstruction. If this body be taken inside, there are at once removed practically all restrictions as to the design of that body. The problem of its design becomes simply that of producing the most efficient high-pressure pump to produce specified results under the conditions of greatly varying load and greatly varying traveling speed of ship.

This leads us back ultimately to the idea of the jet-propeller, and it may be said that this idea has long since been tried and failed; also that the experiment of the propeller set in a fore-and-aft tunnel along the keel of the ship has been tried and failed.

But there may be little hesitation in declaring that these systems were never subjected to fair trial; that there was no persistence in trying improved forms after the failure of the first designs. Rankine said nothing against jet-propellers except that Ruthven's was inefficient because of the evident fault, pointed out by Rankine, that the areas of the jets were, in an extremely large ratio, too small. Because two or three small jets necessarily throwing into the wake an excessive quantity of axially-directed kinetic energy were a failure, it hardly follows that the same system otherwise designed must necessarily be bad. The first Laval steam turbine was not a success and the first Parsons turbine was a still greater failure. The component of discharge velocity represents pure unnecessary success of a jet system, especially in view of its influence upon the skin and wave resistance of the

ship, must greatly depend upon the design of the intake —upon its form and upon its position, upon whether one, two, or a dozen intake ports are used; and upon the distribution of a multiple system of ports. By the light of past experience, one is almost tempted to say that it would have been easy to predict that two, or even three, repelled jets could hardly result in efficient working, and that twelve or sixteen skilfully distributed over the cross section of a bluff stern at such spacings and so disposed as to enable them to carry backwards with them induced currents of water not passed through the inside propelling pump would more likely satisfy the general conditions of economy. For example, an elementary examination of Equation II, shows how great would be the advantage of being able to increase A along with the traveling speed of V, a control easily effected by opening more or fewer of the valves to the different jet tubes or groups of these, according to varying conditions of speed, head wind, etc. With a single, or even a duplex, intake placed far forward, the steering qualities of the vessel are naturally much interfered with, as experience proved if memory serves aright; but with a multiplied intake, with the ports scientifically placed and distributed and controlled by a series of valves equal in number to the ports or proper groups of ports, the stoppage or reversal of the flow through one or several groups of intake ports and stern jets would provide a graduated and powerful method of steering in substitution of, or in reinforcement of, the rudder steering.

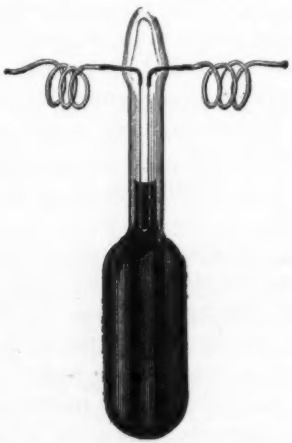
The elementary advantages of jet propulsion have often been discussed, and the object of this paper is not to go over again old ground, although mention should be made of the incidental gain in passing through the hold an enormous flow which may be used as condensation water. The object of the paper is to urge that the scientific mechanics of ship propulsion clearly points towards some form of jet propulsion, and that different forms of this mode of propulsion have never been sufficiently experimented upon. Steam turbines, combined with high-lift compound centrifugal pumps and multiple

intakes and stern jets skilfully distributed in groups, seems to be a solution of the problem of economic propulsion well worth working at.—"The Marine Review."

A THERMO-ELECTRIC PYROMETER OF LOW RESISTANCE AND COMPENSATOR.

This improved electric device for reading high temperatures has been devised by Prof. William H. Bristol, of Stevens Institute. It is designed for use in all kinds of commercial work. Prof. Bristol states that it is similar in principle to the Le Chatelier pyrometer, but it is of low resistance, and instead of the extremely delicate suspension galvanometer, a Weston special dead-beat milli-voltmeter is used. In place of the costly platinum-rhodium elements, inexpensive alloys are employed for the couples.

The low cost of the couple makes it possible to keep an extra one on hand for use as a standard for quickly and easily checking the one that is in regular service.



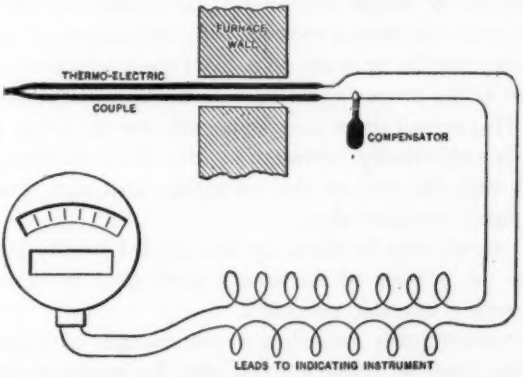
TEMRMO-ELECTRIC COMPENSATOR.

The temperature at a number of localities may readily be observed on a single instrument, a couple and leads being

provided for each locality in connection with a suitable switching device.

The same instrument may also be provided with scales for different total ranges.

For ranges of temperature up to 2,000 degrees Fahrenheit, instead of using porcelain tubes for insulation, each element of the couple is insulated with asbestos and a carborundum paint. Couples so insulated may be applied directly to the fire space where the temperature is to be measured, or, where extra protection is desirable, the couple may be slipped into a piece of common iron pipe with one end closed. Couples so protected are well adapted for use in liquids and molten baths, such as are employed for hardening and tempering of steel.



THERMO-ELECTRIC PYROMETER, COMPENSATOR AND VOLTMETER.

For instantaneous determination of the temperatures of molten metals, as brass, bronze, etc., the ends of the couple are left disconnected and without insulation.

The same form of the couple may be used for quickly measuring the temperature of a metallic object. For this application the tips of the couple are pointed, so that the temperature at the two points of junction may become the same as that of the object immediately after the contact is made. This form of the couple affords a most convenient method for almost simultaneously measuring the temperature at different points of an object.

When desirable the couples are made with a separable junction, which permits the fire end to be removed and renewed at pleasure.

A compensator is adapted automatically to correct for atmospheric changes of temperature at the cold ends of the couple.

In order to make an equivalent and to reduce the cost of the platinum-rhodium couple for the measurement of temperatures above the fusing point of the low-priced alloys, a compound couple is formed with platinum-rhodium for the part to be exposed to the full temperature to be measured and of a length extending to a point where the temperature will not exceed 1,200 degrees Fahrenheit. The remaining portion of the elements of the couple is composed of inexpensive alloys.

Automatic continuous records of the indications of the pyrometer may readily be made on a chart sheet which is arranged to move at the proper speed back to the end of the indicating arm. This record sheet is unsupported over its active portion, which is periodically vibrated by the clock movement into contact with the end of the indicating arm and produces a record upon the chart sheet.

The record may be made by ink carried by the indicating arm, or the surface of the record sheet may be coated with some easily removable substance.

For automatically recording rapid changes of temperature a current from an induction coil may be passed through the record sheet from the end of the indicating arm at frequent intervals.

The idea of the pyrometer is, when heated, the generation of a weak current of electricity, which passes through a compensator on its way to the milli-voltmeter also in the circuit, the degree of heat being indicated on the voltmeter. The compensator equalizes what is termed the cool ends of the pyrometer away from the heat, and keeps their potential of resistance equal by the expansion upward or falling, as the case may be, of a column of mercury, which short-circuits the loop of wire passing down through it. The device avoids the

need of making allowances for the cool end variations, and is very simple and effectual.

The illustration shows the thermo-couple inserted in a furnace, the position of the compensator and the circuit leading to the voltmeter. The apparatus indicates perfectly the slightest changes of temperature.—"Scientific American."

SHIPS.

UNITED STATES.

New Jersey.—This vessel had her official trials March 28-29, and has been preliminarily accepted. On the four hours' trial a mean speed of 19.18 knots was obtained, the main engines developing 22,805 I.H.P. A detailed description of this vessel and her trials will be given in the next issue of the JOURNAL.

Washington.—The official trials of this vessel were held on the Rockland course April 10-11. The mean speed on the four hours' trial was 22.27 knots, the main engines developing 26,862 I.H.P. Full descriptions of the vessel and her trials will appear in the August number of the JOURNAL.

St. Louis.—This vessel is now ready for her official trials, which will be held this month over the Rockland course.

South Dakota.—This vessel underwent satisfactory dock trials at the works of the contractors, the Union Iron Works, San Francisco, Cal., March 20-27, 1906. The official trials will be held in the Santa Barbara Channel in June.

New Revenue Steamer.—The new revenue steamer to replace the *Levi Woodbury* on the coast of Maine, for which Congress has appropriated $200,000, the hull of which is now nearing completion at Rodermond's yard, Tompkins Cove, N. Y., is to be a composite vessel very different in design from other vessels in the service, being intended as an effective ice breaker and to be fitted in other respects for work on this dangerous coast in all weathers. The bow, instead of being ram-shaped, as has been customary on other revenue steamers, is cut away under water to facilitate getting through the ice by breaking it down rather than cutting through it. Although wood is used for the greater portion of the vessel, metal has been worked in wherever it will add to her lightness and

strength without detracting from the object for which the wooden construction was adopted. All deck beams will be of steel, as will also the hanging knees, deck stringer plates, hatch coamings, etc. The sternpost, rudderpost and rudder are to be of manganese-bronze.

The vessel will have three decks extending the whole length, the spar deck being flush forward and aft. The rig will consist simply of two pole masts upon which staysails can be set for steadying the ship in a beam wind. The living quarters will be very commodious, provision being made for a crew of 70 men and 8 officers. In addition to being fitted for carrying a light battery, she is also to be fitted for carrying six four-inch breech-loading rifles. For assistance to merchant vessels large towing bitts and a towing machine are to be provided, as will also complete apparatus for destroying derelicts. The following will be the principal dimensions of the new craft: Length over all, 210 feet; length between perpendiculars, 188 feet; beam, extreme, 36 feet; depth molded, 26 feet 3 inches; displacement to 12-foot 6-inch water line, 1,190 tons.

The steam machinery will consist of a triple-expansion engine of the vertical, direct-acting inverted type, having steam cylinders 20¾ inches, 32 inches and 50 inches in diameter, respectively, by 27-inch stroke, built by the Pusey & Jones Co., Wilmington, Del. She will be fitted with a steam windlass, steam steering engine, steam winch on after deck; an electric generator of 10-kw. capacity, which, in addition to lighting the ship throughout, will be capable of operating a 24-inch search-light located on a platform just forward of the mainmast, and an evaporating apparatus having a capacity of 4,000 gallons of water in 24 hours. The main engine at full power is expected to develop about 1,800 I.H.P., which will drive the ship at a speed of from 14½ to 15½ knots per hour.—"The Nautical Gazette."

ENGLAND.

New Construction.—Between 1st April, 1905, and 31st March, 1906, the following ships will have been completed and become available for service.

3 battleships—*Dominion, Hindustan, New Zealand.*

8 armored cruisers—*Antrim, Argyll, Carnarvon, Devonshire, Hampshire, Roxburgh, Black Prince, Duke of Edinburgh.*

1 second-class cruiser—*Encounter.*

8 scouts—*Adventure, Attentive, Forward, Foresight, Pathfinder, Patrol, Sentinel, Skirmisher.*

16 destroyers.

13 submarines.

1 floating coal depôt.

On 1st April, 1906, there will be under construction—

6 battleships.

10 armored cruisers.

12 destroyers (coastal).

5 destroyers (ocean-going).

1 destroyer (very fast ocean-going).

1 Royal yacht.

15 submarines and a repair ship.

It is proposed to begin during the financial year, 1906-7—

4 armored vessels.

5 destroyers (ocean-going).

12 destroyers (coastal).

12 submarines.

Battleships.—The Committee on Designs, mentioned in last year's statement, considered the various designs, and settled the type of battleship to be laid down during the financial year, 1905-6. This battleship, *Dreadnought,* is being supplied with turbine machinery on the Parsons system, and was laid down at Portsmouth on the 2d October, 1905, and launched on the 10th February by His Majesty the King. It is hoped that the ship will be put into commission in the beginning of 1907.

The *Lord Nelson* and *Agamemnon,* of the 1904-5 programme, which are now under construction on the Tyne and Clyde respectively, have made considerable progress during the year, and it is expected that they will be completed by their contract dates.

The decision to re-arrange the armament in the *Warrior* class has somewhat delayed their completion.

Armored Cruisers.—The six armored cruisers of the *Devonshire* class have passed successfully through their trials, and all of them are in commission. They attained an average speed of over 23 knots, and one reached 23.6 knots. They are practically identical in speed and coal endurance with the cruisers of the *Monmouth* class, but are superior in armament and protection.

The *Duke of Edinburgh* has successfully passed through all her trials, and attained a speed of 22.84 knots in rough water and a strong wind on the measured course at Polperro. She has been completed at Pembroke, and was commissioned with a nucleus crew on 20th January, 1906.

The *Black Prince* has also passed through all her principal trials successfully, attaining a speed of 23.65 knots in fine weather on the Polperro course.

The four later ships of the *Duke of Edinburgh* class, which have an auxiliary armament of 7.5-inch instead of 6-inch guns, have all been launched during 1905-6.

The three armored cruisers of the *Minotaur* class, which were laid down during January and February of 1905, have made good progress.

During the current financial year, contracts have been entered into for the construction of three armored cruisers, named *Invincible, Inflexible, Indomitable.* These vessels are to be ready for commission in May, 1908, *i. e.,* within 30 months from the date of ordering.

Scouts.—All the eight vessels of the new "Scout" class have been satisfactorily completed and put into commission. These vessels have fulfilled all the conditions of the designs, and have obtained speeds for 6½ hours' continuous steaming, varying from 25.06 knots to 25.88 knots. The eight vessels have been built from four different designs.

Destroyers.—All the vessels of the "River" class which were under construction at the commencement of the current financial year have been completed and put into commission, making in all 34 vessels of this class now in service.

The strength and sea-going capabilities of this class have been severely tested, and have proved very satisfactory.

The twelve coastal destroyers and the five ocean-going destroyers included in the current year's programme have all been ordered. The former will have a trial speed of 26 knots and the latter of 33 knots. Negotiations in connection with ordering one special ocean-going destroyer of 36 knots trial speed are nearly complete.

The whole of the destroyers included in this year's programme will have turbine machinery, and be fitted for carrying and burning oil fuel.

Other Vessels.—The design of a new yacht for His Majesty has been completed, and the actual building is in hand on the Clyde.

The steamship *Indrabarah,* now named *Cyclops,* has been purchased for conversion into a fleet repair ship.

Submarines.—The 11 boats in the present year's programme have been ordered.

General.—Improved appliances for cooking, as well as bread bakeries, are being introduced into ships.

Improved sanitary, ventilating, warming and washing arrangements are also being introduced.

The use of electricity for many purposes on board H.M. ships continues to increase.

Machinery and Boilers.—The policy of fitting complete installations of water-tube boilers in war ships has been continued, large-tube boilers of Babcock & Wilcox or Yarrow type being fitted in battleships and first-class cruisers, and boilers of the small-tube type being fitted in scouts and destroyers.

The conditions adopted last year, under which all contractors' machinery trials in new ships were to be carried out under Service conditions, have been applied to all later ships.

The policy of providing reserve sets of auxiliary machinery, etc., has been continued during the past year by the provision of further auxiliary machinery, etc., typical for various classes of ships, and the stock is now practically complete to date. These reserve parts will enable ships to be supplied with replace auxiliary machinery with the least possible delay.

The provision of an adequate supply of fresh water for ships

of H.M. fleet has been under continued consideration. Additional evaporating and distilling plant has been supplied to each battleship which was below latest standard in this respect, when coming in hand for large refit.

All new ships are to be supplied with independent machines for ice making and for cold storage.

Standardization.—The extended policy adopted last year of making the main and auxiliary machinery of ships of the same class interchangeable as far as possible has been continued, and this plan is being carried out in all later ships, including destroyers.

Electric Generating Machinery.—In consequence of the extension of the application of electric motors for driving auxiliary machinery and for ships' purposes generally where applicable, larger installations of generating machinery are being fitted in vessels under construction, and those on the more modern completed vessels are being increased as the ships are taken in hand for large refit.

Turbine Propelling Machinery.—In view of the satisfactory performances of H.M.S. *Ameythyst,* fitted with turbine propelling machinery, and of a rapidly increasing number of such installations in ships of the mercantile marine, it has been decided to adopt this means of propulsion in all the war vessels provided for during the present year.

As the result of experience, it has been decided to replace the set of reciprocating machinery fitted in the *Velox* (turbine-propelled torpedo-boat destroyer) for use at low speeds, by turbines adapted for cruising powers.

Liquid Fuel.—The experimental oil-fuel establishment at Haslar has been completed, and trials with five of the latest types of water-tube boilers, together with instructional work, are in regular progress there.

The oil-fuel installations in the *Mars* and the *Hannibal* are being brought up to date, and, as opportunity affords, oil-fuel appliances are being fitted to H.M. ships *Caesar, Majestic Magnificent* and *Victorious.* Installations are also in progress for all the later vessels building and completing.

The torpedo-boat destroyer *Spiteful,* fitted to burn oil fuel only, is in commission as an instructional vessel for the training of engine-room complements in the manipulation of oil-burning appliances.

Fatality on British Warship.—On the evening of April 16, during the full-power trials of H.M.S. *Prince of Wales* in the Mediterranean, a bolt at the big end of the high-pressure connecting rod of the port engine broke, with the result that the top cylinder cover was fractured, and several stokers were scalded by the escaping high-pressure steam—three fatally.

The Cruiser Duke of Edinburgh has completed her continuous steaming trials, having been twenty-six hours under way, during twenty-four of which coal was burnt in the boilers, and oil fuel for the remainder of the time. The test with oil fuel is a condition recently imposed by the Admiralty, and in the case of the *Duke of Edinburgh* it is reported to have given reasonable satisfaction. The speed of the ship varied between eleven and seventeen knots, and as a result of the trial, the propelling and other machinery will be taken over by the Admiralty from Messrs. Hawthorn, Leslie & Co.

The New Scouts.*—At the Spring Meetings in 1901 I had the honor of laying before the Institution a design for a fast scout, as a probably indispensable adjunct to our battle squadrons. The general ideas as to size, speed, armament and radius of action were my own, the details of the design being prepared for me by the kindness of Sir Philip Watts, who showed what could be done on a given tonnage with the up-to-date knowledge of five years ago. Since that date eight scouts have been built for the Royal Navy, all of them being about 900 tons less displacement, of the same speed, lighter armament, smaller radius of action, but apparently approaching very close to the estimated cost of the above-mentioned design. It has occurred to me that it might be of interest to the members of the Institution if I could collect and lay before them such information concerning the new scouts as might not

*By Admiral C. C. P. FitzGerald, Associate Member of Council. Paper read at the Institution of Naval Architects, April 4, 1906.

fall within the category of "secret and confidential," and the publication of which should not be in any way inimical to national interests.

The scouts have been built in pairs, the orders being given to four firms (two each), and a considerable amount of latitude as to design was allowed by the Admiralty to the builders, as to size, horsepower, and other details, so long as the following conditions were fulfilled:

Speed.—25 knots on an eight hours' trial.

Radius of Action.—Not less than 3,000 nautical miles at 10 knots' speed.

Armament.—Ten 12-pounders, eight 3-pounders, and two 18-inch torpedo tubes, above water.

Protection.—Sloping protective deck, not less than 1½ inches thick, or an equivalent in side armor.

Table I.—THE NEW SCOUTS.

Name.	*Adventure.*	*Forward.*	*Patrol.*	*Sentinel.*	Remarks.
Builder	Armstrong.	Fairfield.	Laird.	Vickers	
Length between perpendiculars, feet	374	365	370	360	*Adventure* longest.
Breadth, extreme, feet and inches	38–3	39–2	38–9	40	
Draught, ft. and ins.	12–3	14–2	14–2	14–1	*Adventure* nearly 2 feet less draught.
Displacement, tons, (Navy List)	2,640	2,860	2,900	2,880	*Adventure* smaller.
Depth, molded, ft. and in.	23	23	23–11	24–6	
Boiler rooms	3	3	4	3	
Boilers	12 modified Yarrow.	12 Thornycroft.	12 Laird-Normand	12 Vickers Express	
Grate area, square feet	784	752	656	716	
Heating surface, sq. ft.	42,686	42,960	43,800	44,053	
Working pressure, lbs.	250	260	275	280	
H.P. cylinders, inches	4, of 22	2, of 31½	2, of 32½	2, of 32½	
I.P. cylinders, inches	4, of 34	2, of 50½	2, of 51½	2, of 52	
L.P. cylinders, inches	4, of 53	4, of 57	4, of 58	4, of 61	
Length of stroke, ins.	25½	30	30	30	
Bunker capacity, tons	450	500	600	410	Considerable difference.

PROTECTION.

Adventure (Armstrong).—Protective deck, 2 inches on slope, ½ inch on flat, covering machinery and magazine spaces, tapering forward and aft. No side armor.

Forward (Fairfield).—Protective deck, 1½ inch on slope, ½ inch on flat, before and abaft machinery spaces. Side armor, 2 inches K.N.C. steel, protecting machinery spaces.

Patrol (Laird).—Protective deck, 1½ inches on slope, ½ inch on flat over boiler and magazine spaces, tapering forward and aft. Side armor, 2 inches K.N.C. steel, protecting the engines.

Sentinel (Vickers).—Protective deck, 1½ inches on slope, ½ inch on flat, all fore and aft. No side armor.

The four firms whose tenders were accepted for the four pairs of scouts were: Sir W. G. Armstrong, Whitworth & Co., Limited, Newcastle; the Fairfield Shipbuilding and Engineering Company, Limited, Govan; Messrs. Cammel, Laird & Co., Limited, Birkenhead; Messrs. Vickers Sons & Maxim, Limited, Barrow. Messrs. Armstrong built the *Adventure* and *Attentive;* the Fairfield Company the *Forward* and *Foresight;* Messrs. Laird the *Pathfinder* and *Patrol;* Messrs Vickers the *Sentinel* and *Skirmisher.* And as the pairs are in all cases sister ships, it will only be necessary to mention one of each pair in the tables given above.

I do not propose to discuss the relative merits of side armor *versus* deck armor. There are high authorities in favor of both methods of protection. Internal sloping armor undoubtedly gives better protection to machinery and boilers than the same weight of vertical side armor; though there seems to be a prejudice afloat against letting shot—and particularly shell—get inside the ship at all. I have not seen any reliable reports from the experiences of the late war which would help to solve the problem, though possibly some such exist.

At their official eight-hours full-speed trials the scouts achieved the following speeds; and, as the twins in this respect were not all alike, for the sake of comparison I give them all. (See Table II.) The weather is not stated, though it was probably one of the causes of the differences in the results. All eight vessels obtained their contract speed, but some of them with very little to spare. It will be noticed that the Elswick twins differ considerably in some respects from the other six vessels. They got the highest speed on trial. They are, roughly speaking, 200 tons smaller than the others. They are

longer and narrower, and draw 2 feet less water. They have six-cylinder engines, with a shorter stroke and higher number of revolutions, and they have no vertical armor; though in this respect they are on a par with the two Vickers vessels. Notwithstanding that the official tonnage of the Elswick twins is about 200 tons less than that of the other scouts, their displacement at the full-speed trials appears to have approached closely to that of the others. They are given as follows: *Adventure,* 2,725 tons; *Forward,* 2,780 tons; *Pathfinder,* 2,808 tons; *Sentinel,* 2,858 tons. The shallow draught of the Elswick vessels might, under certain circumstances, prove to be a great advantage; though, on the other hand, a long, shallow vessel is more likely to pitch her screws out of water than a deeper one; and there is a prejudice afloat against short strokes and quick-running engines.

General Opinions on Scouting.—There seems to be an almost unanimous opinion amongst naval officers that efficient scouting will be of the highest importance in future naval wars; though, on the other hand, there is a wide diversity of views as to what class and size of vessel will be most suitable for the purpose.

Table II.—RESULTS OF TRIALS.

Name.	Speed. Knots.	Indicated horsepower.	Revolutions.	Air pressure in stokehold.
Adventure	25.42	15,850	251.6	2.3
Attentive	25.88	16,212	257.3	
Forward	25.16	15,018	211	1.8
Foresight	25.12	14,277	210	
Pathfinder	25.34	17,176	220	2.6
Patrol	25.06	16,433	213	
Sentinel	25.07	17,488	204	2.7
Skirmisher	25.19	17,031	202	

Some distinguished officers have their eye on the Atlantic liners, and think that they would make better scouts than smaller vessels built for the purpose, on account of their high speed in all weathers, and their great coal capacity. But against this it is pointed out that they afford an immense target for either gun or torpedo; that their machinery is greatly exposed; that they are very costly; that there are not enough of

them of high speed under the British flag; and, finally, that they would in all probability be wanted for other and even more important purposes in case of a great naval war.

Another school of thought seems to indicate large and powerful armored cruisers as the proper vessels for scouting, on the principle that a reconnaissance, to be of any use, must be pushed home in spite of opposition, and that no vessel without armor could do this. There appears to be some force in this argument, though a reconnaissance in force would be an operation of a different character from ordinary inshore scouting by small vessels, which could, if successful, return with the news, or transmit it direct or through an intermediate scout by wireless telegraphy. Armored cruisers are somewhat costly vessels to be risked on scouting work, and they would probably be wanted for other duties.

It is reported that the Japanese used their destroyers to a large extent for scouting purposes, and that they found them satisfactory. I have no direct or very reliable evidence on this point; but even if they did so use them, it does not follow that they are the best class of vessel for scouting. The Japanese probably used them because they were the only available vessels they had which were at all suitable for the purpose. Moreover, in gathering naval lessons from the Russo-Japanese war, it is necessary to bear in mind the peculiar topographical conditions of the area of operations. The Japanese base at Elliot Islands was only about 70 miles from Port Arthur, so that scouts would not require a very wide radius of action during that particular phase of the war; though the advent of the Russian Baltic Fleet probably required scouting over a more extended area, and with larger vessels. Another topographical feature of the Port Arthur campaign was the shallowness of the sea in this district, even far out from the land; and this, taken in conjunction with the absence of strong tides, made submarine mining on an extensive scale remarkably easy, and thus rendered scouting in large and deep vessels specially dangerous. On the whole, the results of the late war appear to have caused a slump in locomotive torpedoes, while there has been a brisk rise in submarine mines.

Sea-going Qualities.—All the new scouts have high forecastles; and, so far as they have been tried, they are said to be excellent seaboats, capable of steaming against a heavy sea, and maintaining a good speed under all ordinary conditions. As a practical illustration of this, I may mention that one of the Laird twins left Gibraltar in company with the fleet a short time ago. All the ships started on their quarterly full-power trial, and in three hours the scout had run the whole squadron out of sight, including the *Amethyst* (turbine third-class cruiser). The scout attained a speed of nearly 25 knots with nine-tenths of her horsepower, a dirty bottom, full up with stores and coal, and deeper than her contract trial draught; and after the four hours' trial was over she maintained a comfortable 18 knots, except in the Bay of Biscay, where she met what is described as three-quarters of a gale of wind, with a heavy head sea, against which her speed never fell below 13.5 knots; and on arrival at Chatham she had enough coal left to steam another 1,000 miles at 10 knots.

There appears to be no difference of opinion as to the ability of the scouts to run down all destroyers in all weathers, with the exception, perhaps, of the 30-knot boats in a flat calm. This has been proved in practice on more than one occasion. They can get 22 knots with half boilerpower; and with their armament of ten 12-pounders (to be increased to fourteen in lieu of the 3-pounders) they ought to make short work of half a dozen enemy's destroyers, the scouts having their vitals protected, and affording a steadier gun platform; though, on the other hand, it is pointed out that they are themselves liable to torpedo attack, almost as much so as battleships, except that their high speed might make it more difficult to hit them.

Comparison with the 1901 Design.—It may not be inappropriate to compare briefly the eight scouts, as built, with the design worked out by Sir Philip Watts in 1901. The latter vessel had a length of 400 feet between perpendiculars, with a beam of 44 feet, and was about 900 tons larger than the existing scouts, though her estimated cost was practically about the same. Her horsepower was about the same, and she was de-

signed for a speed of 25 knots with 500 tons of coal in her bunkers. But the fundamental difference between her and the scouts as built, lies in the fact that she had bunker capacity for 1,200 tons of coal, although, of course, she would not have made 25 knots when so loaded.

This principle, of ships having a legend coal supply with a draught of water at which they can attain their contract speed, and a "full up" supply, at which draught they cannot attain their full speed on first going to sea, is one which I believe is generally accepted and approved. It certainly appears to be reasonable and practical in view of the great importance of a wide radius of action. With 1,200 tons of coal in her bunkers, the 1901 design would have had a radius of action of 8,500 miles at 10 knots. This is more than double the endurance of the existing scouts; and, as the principal complaint against the latter is their very limited coal supply, there are, no doubt, some officers who would prefer the earlier design, even with a slight reduction of speed on first going to sea; but with the certainty that the heavier ship would maintain her speed better in rough water.

Armament.—The 1901 design was intended to carry six 4-inch guns, protected by 4-inch armor, and a dozen machine-guns, unprotected. The armament (revised) of the scouts, as built, is fourteen 12-pounders and two 18-inch torpedo tubes, above water. There seems to be still some difference of opinion as to which of these two armaments is the most suitable for a scout; though I am bound to admit that there is a great preponderance in favor of the 12-pounder armament.

Class.—The scouts form a class of their own, intermediate between cruisers and destroyers, designed to perform special and important duties, which, it is assumed, could not be properly performed by either of the other classes. But whether or no (in the words of the popular phrase) they are destined to "supply a longfelt want" is a question which must be decided by the officers in command of our sea-going squadrons.

It is the opinion of some high authorities that they are not worth the money spent upon them; and, in this connection, it

may be interesting to compare them with the class immediately above them, and the class next below them.

The *Sapphire* and *Amethyst* are our two latest third-class cruisers; their official tonnage is 3,000 (very little bigger than the scouts). The *Sapphire* made 22.4 knots on her trial, and cost, in round numbers, £230,000. The *Amethyst* (turbine) made 23.6 knots, and cost, in round numbers, £227,000. Both ships are armed with twelve 4-inch guns.

The next class below the scouts (omitting the somewhat obsolete torpedo gunboats) will be the new ocean-going destroyers of 800 tons, all of which are to have turbine engines; they are to have a speed of 33 knots, to cost about £138,000 apiece, and to be armed with three 12-pounders and two torpedo tubes.

In comparing the scouts with the third-class cruisers, it does not seem reasonable to condemn the former on account of their much lighter armament, if their higher speed will enable them to perform some specific and important duty, which the latter could not perform. We do not assess the value of the *Vulcan* (torpedo depôt ship), or the *Maine* (hospital ship), or the *Aquarius* (water carrier) in terms of their armament. High speed costs a great deal of money. Armament is cheaper, but will not take the place of speed.

In comparing the scouts with the new ocean-going destroyers, there can be no doubt that the latter will attain a much higher speed on trial; but it does seem doubtful whether they will make suitable scouts, or carry fuel enough for the purpose. They are comparatively small vessels; and it is a well-known fact that the smaller the vessel, the greater is the difference between her trial speed and the speed she will be capable of maintaining at sea. And, conversely, the larger the vessel, the more closely will her ocean speed approach to that which she made on trial.

It is scarcely necessary to add that the new scouts are built with very fine lines; and though a personal inspection of the two Birkenhead vessels would lead one to believe that they have no hollow lines, this may, perhaps, be an optical illusion.

The Cruiser Race.—Fuller details of this interesting event are as follows:

The ships taking part were—

Name.	Displacement.	Horsepower.	Boilers.
Drake	14,100	30,000	Belleville.
Cornwall	9,800	22,000	Babcock.
Essex	9,800	22,000	Belleville.
Bedford	9,800	22,000	Belleville.
Berwick	9,800	22,000	Niclausse.
Cumberland	9,800	22,000	Belleville.

All have a designed maximum speed of 23 knots at full power.

Before leaving New York all ships were coaled to the full capacity. As there was not sufficient Welsh coal available, the flagship *Drake,* record holder of all previous races, was ordered to fill up with American coal—a handicap which created great satisfaction in the squadron. The ships steamed at easy speed to off Sandy Hook, where the signal was made "make your way independently to Gibraltar at greatest speed with the coal on board"—2,500 tons in the *Drake,* and 1,600 tons in the other ships.

At 1.30 P.M. on the 20th of November the ships formed in line off Sandy Hook, received the order to start, and up to the evening of the 24th kept practically all together, the positions being:—

	Ship.	Distance astern of next ahead.
1	*Drake*	——
2	*Bedford*	2,000 yards from 1.
3	*Berwick*	2,000 yards from 2.
4	*Cumberland*	1,500 yards from 3.
5	*Cornwall*	12,000 yards from 4.
6	*Essex*	8,000 yards from 5.

At noon on the 25th the positions were as follows:

1	*Drake*	——
2	*Berwick*	5 miles from 1.
3	*Bedford*	

4 *Cumberland*
5 *Essex* } .....All about 10 miles astern.
6 *Cornwall*

At 3 P.M. on the 25th the *Bedford* developed hot bearings in the port engine, and had to stop that engine for a couple of hours, and so got hopelessly left astern.

In the afternoon of the next day, 26th, the order was—

1 *Drake* ..... ——
2 *Berwick*.....3 miles from 1.
3 *Cumberland*
4 *Cornwall* } .....All about 12 miles astern.
5 *Essex*

On the evening of the 26th most ships had used up their normal coal, and great difficulty was experienced in getting at the reserve supply. All officers and deck hands not on duty volunteered for getting coal out of the reserve bunkers.

At daylight on the 27th the positions were—

1 *Drake* ..... ——
2 *Berwick*
3 *Cumberland* } .....3 miles from 1.
4 *Essex*
5 *Cornwall* } .....About 9 miles astern.

In all the ships coal was now getting very short, and everything depended upon getting it to the furnaces. The amount of coal remaining was—

Drake .....305 tons.
Berwick .....256 tons.
Cumberland .....297 tons.
Essex .....317 tons.
Cornwall .....240 tons.

During the 27th the *Cumberland* began to creep steadily up, and in the afternoon the positions were—

1 *Drake* ..... ——
2 *Berwick*.....1 mile from 1.

3 *Cumberland*.....½ mile from 2.
4 *Cornwall* }
5 *Essex* }.....Well astern.

Five minutes after midnight the *Drake* passed Tarifa Point, the *Berwick* being 1,600 yards astern of her, and the *Cumberland* a little astern again. The other two ships were out of sight, having given up the struggle to a certain extent on account of the difficulty of "scraping bunkers for coal dust."

The *Drake's* time for the whole run was 7 days 7 hours 10 minutes, being an average speed of 18.504 knots for the entire trip. For the first few days a much higher rate was maintained, but towards the end a heavy fall in speed brought down the average in all the ships. All the ships burned coal heavily; it was shoveled on without regard for the usual economical rules. On the fourth day out the *Drake* suffered a good deal of trouble from her American coal, which exhibited a poor calorific value compared with Welsh coal.—"The Engineer."

FRANCE.

The six new French Cruisers which are to be laid down this year will, according to the "Temps," have a displacement of 18,000 tons. Their length will be 145 meters (578 feet), their beam 25.65 meters (84 feet), and their maximum draught 8.44 meters (28 feet). The engines are to be of 22,500 horsepower, giving a speed of 19 knots. The radius of activity with a normal supply of coal will be 4,000 miles at ten knots, or 925 miles at the maximum rate of speed, while with the extra supply which it will be possible to carry these distances will be 8,500 miles and 1,900 miles respectively. The armament will consist of four guns of 305 millimeters (12-inch), 12 of 240 (9½-inch), 16 of 75 (3-inch), and eight of 47 (1¾-inch). It will further include two under-water discharge tubes for torpedoes. The complement will comprise 31 officers and 690 men. The cruisers are to be finished in four years.

French Armored Cruiser Ernest Renan.—On April 9th the French armored cruiser *Ernest Renan,* the largest and most powerful yet laid down in France, was launched at St. Nazaire.

The general dimensions and features of this armored cruiser are as follows:

Length at load water line, 157 m. (515 feet, 1 inch.); breadth, extreme, at load water line, 21.36 m. (70 feet, 1 inch); displacement on trials, 13,644 tons; gross draught, fully loaded, 8.18 m. (26 feet, 10 inches); total bunkers capacity, coal, 2,300 tons; coal carried on trials, 1,354 tons; trials at sea for three hours, 23 knots.

The hull is of steel throughout, fitted with docking and bilge keels. It is protected at the water line by a complete belt armor of 3.70 m. (12 feet, 3 inches), in width, having a maximum thickness of 150 mm. (6 inches) for about 90 m. (296 feet) amidships, the thickness being gradually decreased to 100 mm. (4 inches), and 80 mm. (3.2 inches) at the stem and stern. The two strakes of the belt side armor extend from 1.40 m. (4.5 feet) below the water line to 2.30 m (7 feet, 7 inches) above the same, from the lower to the upper protective decks. The heavy belt side-armor strake has a maximum thickness amidships and at the water line of 150 mm. (7 inches), and this thickness is gradually reduced to the lower edge, where it is only 129 mm. (5.2 inches), and from amidships to the ends where the thickness is 100 and 80 mm. (4 inches and 3.2 inches); the second strake is 130 mm. (5.2 inches) thick amidships, and only 80 mm. (3.2 inches) at the stem and stern. Astern the belt side armor extends only a few feet aft of the rudder, and there an athwartship armored bulkhead extends to both sides and from the lower to the upper protective decks. An armor strake extends from the stem to the fore casemates, and reaches from the top of the water line belt to the lower edge of the gun deck, and is 56 mm. (2.2 inches) in thickness.

There are two armored athwartship bulkheads extending from the shell plating at the ends of the casemates to the 194 mm. (7.7-inch) barbettes, and 150 mm. (6 inches) thick throughout. There are two protective decks extending from the stem to the stern, the lower deck being flat amidships, but sloped at the sides throughout and at each end. It will be built

of 38 mm. (1½ inch) plating throughout, with nickel-steel of 46 mm. (1.84 inches) on the flat, and of 63 mm. (2½ inches) on the slopes. The upper protective deck is flat, and will be built of 38 mm. (1½ inch) double plating. The axis of the lower protective deck is about at a level of the load water line; at the side it is at 1.4 m. (4.5 feet) below it, at the level of the lower edge of the belt side armor. The side protection is completed by a cofferdam extending from the upper protective-deck level to the lower protective-deck level, and worked from end to end of the vessel. Behind this cofferdam there is a watertight bulkhead worked from stem to stern; the space between the cofferdam and the watertight bulkhead is used in its upper part as a passage for repairing breaches in the cofferdam, and in its lower part as a waterway for the water coming through the breaches. Teak backing is fitted behind all side armor. The space between the belt side armor, the watertight bulkhead and cofferdam, and the two protective decks, is called *la tranche cellulaire de protection,* its numerous compartments are either empty, or packed with cellulose or other approved water-excluding material, or filled with coal, fresh water, etc. The ammunition passages, funnels, air pipes, etc., going through the *tranche cellulaire* are protected by an annular cofferdam extending from the lower to the upper protective decks. Above the upper protective deck is the "gaillard deck," or gun deck extending from stem to stern; above this deck is the spar deck, extending from the stem to the astern turret. On the spar deck is a large bridge.

The conning tower is on the bridge deck, between the fore turret and the fore mast. The conning tower and shield will have a thickness of 200 mm. (8 inches). An armored tube 890 mm. (35.6 inches) in diameter will extend from the base of the conning tower to the protective deck and will be 125 mm. (5 inches) thick throughout.

There will be forty-two boilers of the Niclausse type, placed in watertight compartments. There will be six funnels, each 21 m. (69 feet) above the grate level, and 6 m. (20 feet) in diameter.

The coal bunkers are to have a maximum capacity of about 2,300 tons. With the ordinary load of 1,354 tons the steaming radius will be at 10 knots 7,500 miles, and with 2,300 tons 12,000 miles. At full speed the steaming radius is to be 1,026 and 1,630 miles respectively with ordinary and maximum bunkerage. A certain quantity of liquid fuel will be shipped in special tanks and in the double bottom.

All the compartments below the upper protective deck, except the coal bunkers, will be provided with forced-draft ventilation, there being a number of blowers arranged for this purpose. Special attention will be given to spaces subject to habitually high temperature, such as the engine rooms, boiler rooms, and dynamo rooms. All the blowers, except the forced-draught blowers, are to be electrically operated.

The electric plant will consist of four dynamos. The armament will consist of four 194 mm., say 7.7-inch, twelve 164-mm., say 6.5-inch breech-loading rapid-fire guns, twenty-four 47-mm., 3-pounder rapid-fire guns and three 37-mm., 1-pounder rapid-fire guns. The battery will be mounted as follows: The 194-mm. guns in pairs in two electrically-controlled, balanced elliptical turrets on the center line, one forward on the spar deck, the other aft on the gun deck. The 164 mm. guns, of which eight are in electrically-controlled, balanced, elliptical turrets on the spar deck side, four on the port and four on the starboard side, two at each end of the superstructure. The 194-mm. barbettes extend from the protective deck to the spar deck for the fore barbette, and to the gun deck for the aft barbette, and consist of 160-mm. (6.4-inch) armor in front, and 120-mm. (4¾-inch) in the rear; the barbettes will not have any special framing, the connection of the armor with the decks being sufficient. The turrets will have a front plate 160 mm. (6.4 inches) thick, rear plate 120 mm. (4¾ inches) thick, and top plate 39 mm. (1.6 inches thick, with the upper tube 80 mm. (3.2 inches) thick, and lower tube of 30 mm. (1.2 inches) in thickness.

The 164 mm. (6.5 inch) barbettes will be 140 mm. (5½ inches) thick in front, and 100 mm. (4 inches) in the rear,

with the upper tube, 70 mm. (2¾ inches) thick, and the lower tube 20-mm. (.8 inch) thick; the turrets will have front plate 140 mm. (5½ inches) thick, the rear plate of 100 mm. (4 inches) and top plate plate 39 mm. (1.6 inches) thick.

The armor casemates extend from the top of the water-line belt to the lower edge of the gun deck for the forward casemates, and are 140 mm. (5½ inches) in thickness; the athwartship bulkheads at the end of these casemates extend from the shell plating to the 194-mm. (7.7-inch) barbettes, and are 164 mm. (6.6 inches) thick throughout. The casemate armor around the 164-mm. (6.5-inch) guns on the gun deck is of nickel-steel, the front plate is 140 mm. (5.6 inches) in thickness; the splinter and inner plates are 60 mm. (2.4 inches) thick, the floor and top plates are 30 mm. (1.2 inches) thick.

The casemate guns are arranged to fire right ahead and right astern respectively. Two 7.7-inch guns and six 6.5-inch guns will fire right ahead or right astern, and four 7.7-inch guns and six 6.5-inch guns will fire on the broadside. There will be six 3-pounder guns on the gun deck, ten of the same size on the spar deck, and eight on the bridges. There will be two submerged torpedo tubes of 18 inches in diameter.

Magazine bulkheads adjacent to heated compartments, such as fire rooms, engine rooms, dynamo rooms, etc., are provided with air spaces. The shell rooms for the 7.7-inch guns are at the foot of the barbettes; the magazines and shell rooms for the 6.5-inch guns and small guns are between the main engine and firerooms compartments, and are so arranged that about one-half of the ammunition will be carried at each end of the ship. The ammunition will be conveyed directly by hoist from the ammunition rooms, or ammunition passages, to the deck where they are required, or as near that as possible. The hoists will be driven by electric motors; for transporting the ammunition, trolleys on rails will be provided in the handling rooms, passages and shell rooms.

There is a lower bridge forward and aft, and a flying bridge forward. On the flying bridge forward and the lower bridge aft there are chart houses. There are steel masts forward and aft,

the fore mast having an upper and lower top; there is one signal mast, also a searchlight platform forward and aft, and a crow's nest on the forward mast only. Two searchlights will be fitted forward on the gun deck, and two others on the main deck aft. The crew will be composed of 31 officers and 643 men. The vessel will, it is said, be delivered to the Navy on the 5th of August, 1908.—"The Engineer."

Leon Gambetta.—Since 1898 there have been completed for the French Navy thirteen armored cruisers, which may be divided into five types: Three units of 7,700 tons; three of 9,500 tons; five of 10,000 tons; one of 11,300 tons; and one of 12,500 tons. In addition to these, there are under construction three additional cruisers of 12,500 tons and three of 13,600 tons. The particulars of these six types of cruisers will be found given in the table, which covers all of the leading items of information with regard to dimensions, power, speed, battery, protection and steaming radius.

The *Leon Gambetta,* which is the first of the 12,500-ton cruisers to be placed in service, made a trial speed of 23.1 knots as an average for three hours, during which time the indicated horsepower averaged 29,008, with a maximum of 30,500. The Admiralty coefficient at this speed figures out as 230. A twenty-four-hour trial at three-fourths the full power resulted in 16,990 indicated horsepower and a speed of 20.4 knots, which corresponds with an Admiralty coefficient of 271. A twenty-four-hour coal-consumption trial gave a speed of 20.1 knots with 16,600 horsepower and an Admiralty coefficient of 265. In this trial the coal consumption per indicated horsepower per hour figured out at 0.772 kilogram, or 1.7 pounds.

This ship has two protective decks, the lower one of which reaches 4 feet 7 inches below the mean load water line, and is connected with the lower edge of the armor belt. The upper protective deck is 7 feet 6 inches above the water line, being at the top of the heavy side armor, which has a width of 12 feet 1 inch and a thickness of 6.7 inches. This thickness is reduced at the lower edge and is also reduced at the ends of

the ship, at which points it is only 3.5 inches. The upper part of the armor belt has a thickness of five inches amidships and 2 inches at the ends. The belt is carried across near the stern by a bulkhead situated abaft the rudder post, and the extreme stern is unprotected by side armor. There are two other armor bulkheads on the ship, one of them being just abaft the forward barbette and the other just before the after barbette. These bulkheads have a thickness of 4.8 inches and extend from main deck to spar deck, thus protecting the 'tween decks from a raking fire. The whole of the belt armor is laid on a backing of teak, and water-excluding material is packed in a cofferdam behind the armor and backed up by a watertight longitudinal bulkhead. The thickness of the lower armor deck varies from 2.2 inches to 1.8 inches, while the upper armor deck has a thickness varying from 1.3 to 0.8 inches.

The battery of this ship includes four guns of 7.7 inches; sixteen of 6.4 inches in caliber; twenty-two 3-pounder rapid-fire guns and five torpedo tubes, of which two are submerged. This is so disposed that the fore-and-aft fire consists in two 7.7-inch guns and six 6.4-inch guns, while there may be concentrated on either broadside four of the former and eight of the latter. The 7.7-inch guns are mounted in pairs in closed barbettes forward and aft, and twelve of the 6.4-inch guns are similarly mounted in pairs in turrets on the spar deck, the other four being in casemates. Of the 3-pounder guns, six are on the upper armored deck, ten on the main deck and six on the bridges.

The protection for the heavy turret guns consists in armor with a thickness of 7.9 and 5.5 inches respectively, while the casemates inclosing four of the 6.4-inch guns have walls of a thickness of 4.7 inches with 1-inch splinter bulkheads, 1-inch ceiling and 1.2-inch floor plates. These casemates are really continuations of the side armor, which makes it impossible for them to be destroyed by cutting away the underpinning, as is the case in many constructions.

The ship is driven by three screws actuated each by a separate four-cylinder, triple-expansion engine with piston diam-

eters 39-inch, 59-inch and two each of 65-inch, and a common stroke of 37 inches, and operating at 125 revolutions per minute. These engines are located amidships between the sets of boilers. The latter, twenty-eight in number, are of the Niclausse type, with a total grate surface of 970 square feet, and operate at a pressure of 216 pounds per square inch. Each set of fourteen boilers is served by two funnels measuring 65 feet above the grate bars. The maximum speed obtained during the trial runs over the measured mile was 24 knots, which required 30,500 horsepower. Several other individual runs go as high as 23.6 knots. The consumption of fuel is such that with the total bunker capacity of 2,100 tons of coal, the steaming radius at 11 knots figures out at 12,000 nautical miles, and 1,500 miles at 22 knots. In addition to the coal supply, provision is made for carrying 100 tons of liquid fuel. The normal bunker capacity is 1,400 tons.

Ventilation of all compartments below the protective deck is taken care of by means of electric fans, fresh air being drawn in through ducts located between the funnels. Bilge pumps are fitted, with a total hourly capacity of 700 tons of water. Electricity is used to a considerable extent not only for lighting but for certain of the auxiliary machinery, turret turning, ammunition hoists, steering gear and fans. There are six searchlights, two being fitted forward and two aft, and one on each mast. Electricity is generated by means of four dynamos, three being required for constant service, while the fourth is in reserve. The ammunition hoists carry ammunition for the 7.7-inch guns from magazines directly beneath the barbettes. Other ammunition is stored in rooms between the main engine rooms and the boiler rooms.

Taking the ship as a whole, she forms an interesting solution of the problem of condensing in a moderate displacement high speed, powerful artillery, good protection and a satisfactory steaming radius. She represents the resultant of improvements indicated by experience with previous types, and has got away from the bad features incident to the early attempt to include much the same military qualities in a vessel of 7,700

tons. In these small ships the engines are fragile, the hull weak and the cost of keeping the ship in good repair overbalances the difference in first cost which would have been required for better and larger constructions.

The *Leon Gambetta* is known as a vanguard cruiser, having all the necessary qualities to search for the enemy and to keep in contact with him; her speed and small artillery insure her against successful attack by destroyers, while her protection and heavy artillery make her able to fight on equal terms against similar cruisers in foreign navies, and momentarily with a battleship. It would be impossible, however, for her to long sustain an engagement with such a powerful antagonist, because of the fact that her superstructure would be destroyed in a very short space of time, and with such effect as probably to nullify all attempts to supply the turrets and guns and keep them in working order. Not only this, but the destruction of the base of the funnels would fill the upper works of the ship with smoke and hydro-carbon gases from the fires, rendering asphyxiation extremely likely. This would also operate to neutralize the effect of forced draft, and thereby cut down the speed.

The second part of the table deals with similar cruisers of about the same sizes in foreign navies. One of these ships, the *Roma,* is usually called a battleship, but her speed is such that she properly comes under the classification known as "battle cruisers" which represents what may be expected to be in some measure the ultimate outcome of the present approachment in type between the battleship and the armored cruiser.—"Marine Engineering."

French Torpedo Boats.—The French shipbuilding firm, Chantiers de la Loire, has received the order to construct six first-class torpedo boats for the French Navy, Nos. *347* to *352*. They are to steam at twenty-six knots, like their predecessors, and to have three torpedo tubes, one in the bows and two aft, but the engine power has been increased from 1,900 to 2,000, and the speed of twenty-six knots will be exceeded, several of the boats of the quota have already steamed at twenty-seven knots.

GERMANY.

The Past Naval Year.—During 1905 the two new first-class battleships *Preussen* and *Hessen*—two of the *Braunschweig* class—each of 13,200 tons displacement, have been completed, commissioned, and taken their place in the second Squadron of the Active Battle Fleet; two first-class armored cruisers, the *Roon* and *Yorck*, of 9,500 tons displacement each, have also been completed; the latter having finished her trials has taken her place as flag ship of the Rear Admiral Commanding the Scouting (Cruiser) Squadron of the Battle Fleet, while the former, as soon as her trials are completed, will become the flag ship of the Commodore Commanding the second Division of the Squadron. In addition, three small cruisers, the *Berlin*, *München* and *Lübeck*, have been completed; of these the first-named has joined the Second Division of the Scouting Squadron, while the other two are still carrying out a series of competitive trials, the *Lübeck* being fitted with turbines.

Approaching completion are the two first-class battleships *Lothringen* (the last of the *Braunschweig* class) and *Deutschland*, of 13,200 tons displacement, both launched in 1904, while two others of the same class, the *Hannover* and *Pommern*, were launched last year. No armored cruiser was launched, but three small cruisers, the *Leipzig*, *Dantzig* and *Königsberg*, took the water. Building and to be launched this year are the two battleships *Q* and *R*, also of 13,200 tons—under construction at the Schichau Yard, Dantzig, and the Germania Yard, Kiel, respectively—which will bring the number of this group of powerful battleships up to ten, the last five, in which the *Deutschland* is included, being slightly superior in some details in their armor protection to their five predecessors of the *Braunschweig* class. There are also building two first-class armored cruisers, *C* and *D*, of 11,500 tons displacement, laid down last year in private yards at Bremen and Hamburg respectively, and three small cruisers, the Ersatz *Meteor*, Ersatz *Wacht* and Ersatz *Blitz*.

It is further proposed to lay down during the current year two more battleships, which are to have, roughly, a displace-

ment of 15,000 tons, which are to replace the old third-class battleships *Baiern* and *Sachsen,* launched as long ago as 1878; a first-class armored cruiser, *E,* of 12,000 tons displacement; two third-class cruisers, and two torpedo-destroyer divisions of six boats each. The two third-class cruisers are to replace the old despatch vessels, *Pfeil* and *Komet,* launched in 1882 and 1892 respectively.

The following table gives a list of the ships completed and under construction last year and to be laid down this:

Name.	Type.	Displacement.	Date when laid down, launched or completed.	Where built or building.
Preussen	Battleship.	13,200	Completed 1905	Stettin
Hessen	"	13,200	" 1905	Kiel
Lothringen	"	13,200	Launched 1904	Dantzig
Deutschland	"	13,200	" 1904	Kiel
Hannover	"	13,200	" Sept. 29, 1905	Stettin
Pommern	"	13,200	" Dec. 2, 1905	Wilhelmshaven
*O**	"	13,200	Begun 1905	Kiel
*R**	"	13,200	" 1905	Dantzig
Ersatz Bayern	"	18,000	To be begun 1906	...
Ersatz Sachsen	"	18,000	" 1906	...
Roon	Armored Cruiser	9,500	Completed 1905	Kiel
Yorck	"	9,500	" 1905	Hamburg
C	"	12,000	Begun 1905	Bremen
*D**	"	12,000	" 1905	Hamburg
E	"	12,000	To be begun 1906	...
Lübeck	Protected Cruiser	3,200	Completed 1905	Bremen
Berlin	"	3,200	" 1905	Dantzig
München	"	3,200	" 1905	Bremen
Leipzig	"	3,200	Launched Mar. 22, 1905	Bremen
Dantzig	"	3,200	" Mar. 22, 1905	Dantzig
Ersatz Meteor	"	3,200	Begun 1903	Kiel
*O**	"	3,200	" 1905	Dantzig
Ersatz Wacht	"	3,200	" 1905	Bremen
Ersatz Blitz	"	3,200	" 1905	Stettin
Ersatz Pfeil	"	3,200	To be begun 1906	...
Ersatz Komet	"	3,200	" 1906	...

* Details uncertain.

The Supplementary Navy Bill laid before the Reichstag last autumn allows for the construction between the present year and 1917 of eighteen battleships, thirteen armored and twenty-four small cruisers, the Navy Law of 1900 having allowed for each of these classes, seventeen, ten and twenty-nine vessels respectively; seventeen of these battleships will replace older

vessels, viz.: The four ships of the *Sachsen* class, the *Oldenburg*, the eight coast-defense vessels of the *Siegfried* class, and the *Brandenburg*, with her three sisters. In addition, the sixteen destroyer divisions of six boats each (ninety-six boats), which are being and to be constructed under the Bill of 1900, are to be increased to twenty-four divisions, making a total of 144 boats.

The building programme, 1906-1917, is to be carried out as follows:

Year of first vote.	Navy Act and Supplementary.			Destroyer divisions.	Cost of construction and armament.	
	Battle-ships.	Cruisers.			Million Marks.	£
		Large.	Small.			
1906	2	1	2	2	117 =	5,728,125
1907	2	1	2	2	131	6,413,542
1908	2	1	2	2	136	6,658,333
1909	2	1	2	2	145	7,098,958
1910	2	1	2	2	148	7,245,832
1911	1	2	2	2	147	7,196,874
1912	1	2	2	2	144	7,050,000
1913	1	1	2	2	137	6,707,291
1914	1	1	2	2	126	6,168,750
1915	1	1	2	2	119	5,826,041
1916	1	1	2	2	112	5,483,334
1917	2	...	2	2	113	5,532,292
	18 Including one still to be built under Act of 1900.	13 Including the six Armored Cruisers under the Supplementary Act of 1906.	24 Including one still to be built under Act of 1900.	24		

—"Etat für die Verwaltung der Kaiserlichen Marine, 1906."

Steam Trials.—The new first-class battleship *Hessen* has concluded her trials satisfactorily. During a twenty-four hours' coal-consumption trial, with the engines developing 11,384 I.H.P., the mean speed was 16.6 knots, with a coal consumption of 0.791 kg. (1.75 pounds) per I.H.P. per hour; at a further trial, with the engines developing 16,900 I.H.P., the speed attained was 18.2 knots. In a six-hours' forced-draft

trial, with the eight water-tube boilers, the engines developed 14,285 I.H.P., and with the six cylindrical ones, 4,364 I.H.P., making 103 and 79.8 revolutions respectively, the coal consumption with the cylindrical boilers being 0.822 kg. (1.81 pounds) per I.H.P. per hour.

The new first-class armored cruiser *Yorck* has also concluded her trials. In her preliminary trials last December in Echenförde Bay, with the engines developing 12,930 I.H.P., and making 99.6 revolutions, with an air pressure of 13.5 mm. (0.53 inch), a mean speed of 19 knots was obtained; under forced draft, with an air pressure of 22-mm. (0.866 inch), the engines developed 19,183 I.H.P., giving a speed of 21.1 knots. At the trial runs for testing the best pitch for the screws, the best results were obtained with a pitch of 6.8 cm., which, with the engines developing 19,044 I.H.P., gave a speed of 20.7 knots, which, when steaming in deep water, will mean a speed of over 21 knots. At the official trials in February during a twenty-four-hours' run with the side screws only, using eight boilers, the engines developed 3,880 I.H.P., making 74.2 revolutions and giving a speed of 12.9 knots, with a coal consumption of 0.86 kg. (1.9 pounds) per I.H.P. per hour. During another twenty-four-hours' run with all boilers alight, the engines developed 13,711 I.H.P., making 103 revolutions, and giving a speed of 19 knots, with a coal consumption of 0.896 kg. (1.97 pounds) per I.H.P. per hour. During a six-hours' run in deep water under forced draft the engines developed 20,295 I.H.P., making 118.6 revolutions, and giving a speed of 21.4 knots; coal consumption not recorded.

The third-class turbine-propelled cruiser *Lübeck* has been continuing her trials. At her last run, using four large and four small propellers, the following results were obtained:

I.H.P.	Revolutions.	Speed.
2,337	353.3	14.65 knots.
3,080	379.7	15.63 knots.
4,003	415.1	17.1 knots.

The six new destroyers of the Series *S120* to *S125* have attained the following mean speeds on the three-hours' acceptance trials under forced draft:

With a displacement of 390 tons:

S120	27.01 knots.
S121	27.09 knots.
S122	27 knots.
S123	28.27 knots.
S124	28.30 knots.

With a displacement of 460 tons:

S125 (turbines)	26.60 knots.

The turbine-engined *S125* has made some further three-hours' forced-draft trials, when she averaged 27.75 and 28 knots; although of a nominal displacement of 460 tons, it is stated that in reality her displacement is no higher than that of the other boats, *S120* to *S124*. The greater speed of *S123* and *S124* over *S120*, *S121* and *S122* is due to improved screws, and that they keep steam better, and certain alterations are accordingly to be made in the latter.

New Ships.—On the 2d December the new first-class battleship *O* was launched from the Vulcan Yard at Stettin, and received the name of *Pommern;* she is the third of the *Deutschland* class to take the water, and her dimensions are as follows: Length, 430 feet 2 inches over all, 307 feet 6 inches between perpendiculars; beam, 72 feet 10 inches; mean draught, 26 feet, with a displacement of 13,200 tons. Protection is afforded by a complete water-line belt of Krupp steel, with a maximum thickness of 9.8 inches amidships, tapering to 5.9 inches and 3.9 inches at the extremities; above the water-line belt is another, 7.9 inches thick, reaching to the main deck, forming a central citadel, some 240 feet long, while on the main deck is a casemate battery for the secondary armament, protected by 6-inch armor; the barbettes for the heavy guns are protected by 11-inch armor, with hoods of the same thickness, and the casemates on the upper deck for four of the 6.7-inch guns are 6-inch; the transverse bulkheads are

6-inch, the foremost conning tower 11.8-inch, and the after one 6-inch. The engines are to develop 16,000 I.H.P., giving a speed of 18 knots, steam being supplied by twelve Schultz-Thornycroft water-tube boilers. The ordinary coal storage will be 800 tons, which can be increased to 1,800 tons, with 200 tons of oil fuel. The armament consists of four 11-inch guns, mounted in barbettes for and aft, fourteen 6.7-inch Q.F. guns, ten in the main battery and four in casemates on the upper deck, twenty-two 3.4-inch guns, with eight machine guns. There are some slight improvements in the ships of this class over those of the *Braunschweig* type; the armor belt is .8 of an inch, the citadel armor 2 inches, and the casemate battery 1 inch thicker than in the earlier ships, while the four 6.7-inch Q.F. guns on the upper deck have been placed in separate casemates instead of in turrets.

The new third-class cruiser Ersatz *Wacht* is to be built at the Vulcan Yard, Stettin, and to be fitted with turbine engines, which will be supplied by the German Turbine Parsons Company. They are to develop 13,600 I.H.P., as against the 11,600 I.H.P. of the *Lübeck,* and are estimated to give her a speed of 24 knots, as against 23 of the last-named ship. Her displacement will be 3,410 tons, against the *Lübeck's* 3,250; her length 354 feet 2 inches; her beam 42 feet 3 inches, and her draught 13 feet 9 inches; she is thus 12 feet longer and 3 inches more beam than the *Lübeck.* It is understood that the dimensions and speed of the two other third-class cruisers, the Ersatz *Pfeil* and the Ersatz *Komet,* will be the same as those of the Ersatz *Wacht,* but they will be fitted with reciprocal engines.

The five protected cruisers of the *Hertha* class are being modernized, at a cost of £50,000 each; among other improvements, they are to be fitted with water-tube boilers. The *Freya* and *Veneta* were taken in hand last year, and work on the *Hertha* and *Victoria* will be commenced as soon as the necessary sum has been voted by the Reichstag. The *Hansa,* the fifth of the class, is at present in China. These vessels have a displacement of 5,800 tons and a speed of 18.5 knots.

Their armament, which consists of two 8.2-inch Q.F. and eight 5.9-inch Q.F. guns, is protected by armor, the 8.2-inch guns in hooded barbettes of 3.9-inch steel, one forward and one aft, and the 5.9-inch Q.F. guns, four in turrets and four in casemates, 3.9-inch thick, with 3.1-inch ammunition hoists. The fore conning tower is protected by 8-inch armor, and the after by 3.9-inch, while there is a funnel glacis 4.7 inches thick.—"Marine Rundschau" and "Neue Preussiche Kreuz Zeitung."

It is now established that the new German battleships will be of 19,000 tons displacement, and will carry fourteen 12-inch guns. The arrangement of the guns, which is of most interest, has not yet been decided.

The German Admiralty have decided upon an improvement in their new torpedo-boat destroyers. These new vessels will exceed in size our River class, being 570 tons displacement, whereas the largest of preceding vessels was of 420 tons. But instead of reducing the speed to 25½ knots, which was done in the British vessels, the Germans propose to increase their rate to 30 knots, as compared with 27 knots in the case of earlier vessels of this type. The boilers and engines will be protected by light armor, probably of hardened nickel-steel, which may keep out shot from small machine guns firing at long range. Even the modern three-pounder gun is able to penetrate only 2 inches of wrought iron at 1,000 yards range, and the twelve-pounder only 5 inches. The armament of these boats will be specially heavy. There will be eight 1.97 and two 3.47-inch guns, whereas in previous destroyers the ordnance was entirely of the former light weapon. The fuel capacity is also being increased to give a greater radius of action, notwithstanding higher speed.—"The Practical Engineer."

Yorck.—This armored cruiser, the last of the *Prinz Heinrich* class, completed her trials in February, with the following results:

	Dec. 15.	Dec. 16.	Feb. 1.	Feb. 7.
Revolutions .	99.6	.....	74.2	118.6
I.H.P.	12,930.0	19,183.0	3,880.0	20,295.0
Speed	19.0	21.1	12.9	21.4

The trial of February 1 was of twenty-four hours' duration, was made with eight of her sixteen boilers, and with only the two lateral engines in use. Before this trial a change in the pitch of the propellers to 22.31 feet was reported, and a speed of 20.7 knots with 19,040 I.H.P. The last trial was made in deep water.

The principal dimensions of the *Yorck* are—

Length on water line,	feet	403.25
Beam	feet	65.5
Draught, mean	feet	24.0
Displacement	tons	9,500.0

The battery comprises four 8.2-inch, ten 5.9-inch, twelve 3.4-inch and fourteen smaller guns and four torpedo tubes, three of which are submerged.

There are three vertical triple-expansion engines, each in a separate watertight compartment. The boilers are of the Dürr type, sixteen in number, the principal particulars of which, as well as for the boilers of the other cruisers of this class are given below:

	Prinz Heinrich.	*Prinz Adelbert.*	*Friedrich Karl.*	*Roon* and *Yorck.*
I.H.P.	15,700	17,600	17,700	19,000
Steam pressure	213	192	202	206
Grate surface	1,017	1,084	1,084	1,250
Heating surface	45,177	45,963	47,040	52,788
Weight of boilers and water	.....	541.8*	551.7*	607.8*

* Metric tons.

JAPAN.

From the papers just to hand from Japan we learn that the new ships under construction, or to be immediately laid down as additions to the Japanese Navy, aggregate some 97,000 tons. They are as follows:

First-class battleships:—*Aki,* just laid down in the berth hitherto occupied by the *Tsukuba* at Kure; 19,000 tons. *Satsuma,* already under construction at Yokosuka; 19,000 tons.

First-class armored cruisers:—*Tsukuba,* launched recently at Kure; 13,000 tons. *Ikoma,* just launched at Kure; 13,000

tons. *Ibuki,* to be launched in August, at Yokosuka; 13,000 tons. *Kurama,* under construction at Yokosuka; 13,000 tons.

Third-class cruisers:—*Tone,* under construction at the Kawasaki Yard; 2,500 tons. *Yodo,* under construction at the Kawasaki Yard; 2,500 tons. *Mogami,* under construction at the Mitsu Bishi Yard, Nagasaki; 2,500 tons.

Total:—Two battleships, four first-class armored cruisers, and four third-class cruisers; aggregating 97,500 tons.

Kashima.—The two battleships simultaneously ordered early in 1904 by the Japanese Government—one from Sir W. G. Armstrong, Whitworth & Co., Limited, of Elswick, and now known as the *Kashima,* and the other from Messrs. Vickers Sons & Maxim, Limited, and called the *Katori*—have progressed side by side. They will form a most powerful addition to the war-worn fleet of the Mikado, representing, if not the last word in battleship design, at all events a splendid advance on even the ships of our *King Edward VII* class.

The *Kashima* has a length between perpendiculars of 425 feet, a beam of 78 feet, and a depth, molded, of 43 feet 6 inches; with a mean draught of 26 feet 7½ inches, the displacement is 16,400 tons. At this displacement she carries 750 tons of fuel, but capacity is provided for 2,150 tons. The machinery, which was constructed by Messrs. Humphrey, Tennant & Co., Limited, of London, was designed to give the ship a speed of 18½ knots.

The armament, which is of most interest at the present time, is an improvement upon that in the *King Edward VII* class. The vessels were ordered before the war, so that they do not embody such lessons as the operations may have suggested to the Japanese; they are rather developments of British designs. There are four 12-inch breech-loading guns, two mounted in a barbette forward and two aft, while at the four corners of the citadel there are separate 10-inch guns in barbettes. Amidships on the upper deck there is a 6-inch gun on each side, and within a box battery on the main deck five 6-inch guns on each side, making twelve in all. This, it will be recognized, is a more powerful armament than in vessels of the *King Ed-*

ward VII class, which have only four 12-inch, four 9.2-inch, and ten 6-inch guns, and, moreover, each weapon of the Japanese ship is of greater power. In the later ships the Japanese have dispensed with the 6-inch gun entirely, and, in addition to four 12-inch guns, have a secondary battery of 10-inch weapons. The machine guns of the *Kashima* include twelve 12-pounder quick-firing guns, three 3-pounders, and six Maxim guns, while, for firing 18-inch torpedoes, there are five submerged torpedo tubes, which, during the trials, proved efficient when fired with the vessel steaming at 15 and 17 knots.

As to the armor, the water-line belt in the center part of the ship is 9 inches thick, tapering by easy stages to 4 inches at the bow and stern. The next strake, protecting the citadel containing the 6-inch guns, is of 6 inches thickness, and the armor for the upper deck battery is of 4 inches. The 12-inch gun barbettes are of 9-inch armor for the most part, and 5 inches where the walls are otherwise protected by bulkheads or broadside armor. The 10-inch gun barbettes are of 6-inch, reduced to 2-inch behind the bulkheads. The conning tower has 9-inch armor, and the observing tower, which is located above the conning tower, 5-inch armor.

The triple-expansion engines have cylinders which are respectively 36 inches, 56 inches and 63 inches in diameter, with a 48-inch stroke. The design follows generally Messrs. Humphrey, Tennant & Co.'s practice, which we illustrated fully in connection with one of the later British warships (see "Engineering," vol. lxxix, page 340). Steam is supplied by twenty water-tube boilers of the latest design of the Niclausse type, the total heating surface being 42,960 square feet, while the working pressure is 430 pounds. The heating surface is therefore equal to 2.23 square feet per unit of power developed. With the exception of the machinery, the whole of the ship, including armor and armament, electric installations, fittings, &c., has been constructed by Sir W. G. Armstrong, Whitworth & Co.

The trials were carried through in a remarkably short period of time. They commenced with a preliminary run on April

3d, and were completed in six days. They were conducted in the North Sea, the measured runs being made over the Admiralty course off the mouth of the Tyne. In the first place, a series of progressive speed trials were made over this course, with the average results tabulated below; to these we have added speed results of the twenty-four-hours' trial, and the eight-hours' full-power trial, in order to complete the progressive speeds of the vessel.

Revolutions per minute.	I.H.P.	Speed, knots.
69.5	3,030	11.136
89	6,275	14.27
102	9,160	16.323
110	11,400	17.204
113.6	13,000	18.
123	17,280	19.242

The results at 18 knots represent the mean performance during the twenty-four-hours' trial at four-fifths full power, which is supposed to give the continuous steaming performance of the ship, while the details for the 19.242 knots are the average results attained during four runs over the measured course, while the vessel was running her full-power trial of eight-hours' duration. Following upon the four hours, she continued at the same revolutions until the end of the sixth hour, when the engines were speeded up to nearly 125 revolutions, and this continued for two hours; the average speed during that period was probably 19½ knots. In any case, the speed results are exceptionally favorable; the contract required 18½ knots, with the engines indicating 15,600. The mean for the eight hours was therefore three-quarters of a nautical mile per hour higher for an excess of power of 1,680 indicated horsepower.

As to the coal consumption, the result on the twenty-four-hours' continuous-steaming trial at 18 knots was 1.86 pounds per indicated horsepower per hour, and on the eight-hours' full-power run, 2.12 pounds per indicated horsepower per hour. The results were thoroughly satisfactory. The gun trials were also carried out while the vessel was at sea, three rounds being

fired from each of the large guns, when trained at an angle estimated to impose the most severe test upon the structure of the ship and the gun mountings. The only effect was the usual breaking of some glass and minor fittings. The trials, which were conducted under the direction of Captain Tanaka and Captain Fujii, with their respective staffs, on behalf of the Japanese Admiralty, were concluded on the 9th of April, after which the vessel returned, to be prepared for commission by Captain Ijichi. It was anticipated that this vessel would leave by the end of May, so that from the date of the laying of the keel until the delivery of the ship only two years and two months will have elapsed—a performance worthy of the high traditions of the company, and four months within the contract time.—"Engineering."

The trial results of the *Kashima* offer an interesting comparison with the performance of the *King Edward* type of battleship, because the length and beam are the same and the draught and displacement vary little, the *Kashima* being immersed 1½ inches less, and the displacement is 50 tons less. For 3,030 indicated horsepower the speed was 11.136 knots, for 6,275 indicated horsepower 14.27 knots, for 9,160 indicated horsepower 16.323 knots, for 11,400 indicated horsepower 17.204 knots, for 13,000 indicated horsepower 18 knots, and for 17.280 indicated horsepower 19.242 knots. The designed speed at 16,400 tons displacement was 18½ knots, so that the actual result is practically three-quarters of a mile per hour higher. The British ships had about 12 per cent more heating surface in the boilers, and, therefore, could develop about 1,000 more indicated horsepower; yet the average speed at full power is not greater. The *Hindustan*, by Messrs. John Brown and Company, for 18,521 indicated horsepower, got by the showing of the log 19.8 knots; the *Commonwealth*, by the Fairfield Company, for 18,562 indicated horsepower, 19.37 knots; the *Dominion*, by the Vickers, for 18,438 indicated horsepower, 19.35 knots; while the *New Zealand* on the measured course got only 18.59 knots for

14,440 indicated horsepower. The *King Edward* logged only 19.04 knots. A comparison of the speeds got during the long trial at 13,000 indicated horsepower also shows a superiority for the Japanese ship, and this is likewise obvious at low powers. It would be interesting to have an explanation. It may be (says the "Glasgow Herald") that the experience gained with the screw propellers in the British ships has been successfully utilized. As regards coal consumption, however, our records are more favorable. On the eight-hours' full-power trial the *Kashima* burned 2.12 pounds per indicated horsepower per hour. The rate varied from 1.77 pounds in the case of the Vickers ship to 2.17 pounds in the case of the vessel engined by Harland & Wolff. While on the trial at 13,000 indicated horsepower the rate on the *Kashima* was 1.86 pounds the Vickers and the two Clyde-built ships averaged 1.7 pounds, the Belfast-engined ship 1.96 pounds, and the *New Zealand* 1.83 pounds.

Katori.—Interest in the official trials of the Japanese battleship *Katori,* which terminated on May 1, centered largely in the tests of the guns and gun mountings, as Messrs. Vickers Sons & Maxim, Limited, who designed and constructed the vessel from keel to truck, ready for war service, have embodied in the ordnance many improvements in detail, as a result of experience and experiment. Each gun is more powerful than those fitted in corresponding positions in previous ships built in this country, and collectively they give the vessel a greater power of attack than is possessed by any warship now in commission, while, moreover, the vessel attained the exceptionally high speed of 20.22 knots as a mean of several runs over a long course. For instance, the 9.2-inch guns in the barbettes at the four corners of the citadel in the ships of the *King Edward* class are displaced in the *Katori* by 10-inch guns, which develop 27,570 foot-tons, against 21,150 foot-tons by the smaller breechloaders in the British vessels. The gun trials were directed first to determine the effect of the higher power of these heavier pieces on the structure of the ship, and although the guns were fired at high elevations

and low depressions, and on various bearings throughout the large arc of training, no injury resulted. Another and even more important point was as to the influence on the rate of fire consequent upon the use of the larger projectiles, &c.; it was found that the 10-inch gun fired five rounds in 2 minutes 8.4 seconds, and the 6-inch guns eight shots in 52.4 seconds—both good performances.

The vessel has notably fine lines. The following are the principal dimensions:

Length between perpendiculars, 420 feet; length over all, 455 feet 9 inches; breadth, 78 feet; depth to upper deck, 44 feet; draught, 27 feet; displacement, 15,950 tons; speed on trial, 20.22 knots; normal coal supply, 750 tons; full coal supply, 2,100 tons.

Arrangement of Armor.—The main armor belt has a depth of 7 feet 9 inches, of which 5 feet 3 inches is below the water line, and extends from end to end of the vessel, its thickness being 9 inches for a length of 240 feet amidships, reduced gradually to 4 inches at the stem. The aftermost part of this belt consists of 2½-inch armor secured to 1-inch plating, making a total thickness of 3½ inches. Armor bulkheads, 9 inches thick, are carried across the ship at the forward and after ends of the 9-inch belt, extending in depth from the lower to the middle armored decks. From the main belt to the main deck a secondary armor belt is built, extending from abreast the after 12-inch gun barbette to the stem, its thickness being 6 inches for about 185 feet amidships, reduced to 4 inches at the stem; the main deck from the forward barbette to the stem above this belt is 1 inch thick. At the after end of this belt the protection is completed by an armored bulkhead 6 inches thick, carried across the ship, the center portion, 10 inches thick, forming part of the after barbette. Above the secondary belt, and extending from the forward to the after 12-inch gun barbette, there is 6-inch armor on the side up to the upper deck, carried obliquely across at each end to meet the barbettes, and forming a battery, in which are placed ten of the 6-inch guns, divided

from each other by bulkheads of nickel-steel, and separated from the remainder of the deck space by longitudinal bulkheads of nickel-steel extending for the full length of the battery. The upper deck for the full extent of this battery is formed of steel plates 1 inch thick. Amidships on the upper deck, extending for a length of 74 feet, is a battery 7 feet 6 inches in height, of 4-inch armor, in the center portion of which the remaining two 6-inch guns are placed, the roof of this battery being formed of nickel-steel 1½ inches thick. The armor on the barbettes enclosing the 12-inch guns is 10 inches in thickness generally, but reduced to 5½ inches thick where protected by the belt and battery armor. The 10-inch guns are mounted in barbettes, the upper portion of the armor being 6 inches thick and the lower portion 2 inches; the tubes inclosing the ammunition hoists are also 2 inches in thickness. The armor of the conning tower is 9 inches thick, and the observer tower 5 inches thick, the tubes for communication from these being 8 inches and 4 inches thick respectively. As in recent ships of the British Navy, entrance to the conning tower is from the wheel house through a hatchway in the armored roof. The protective deck is of a minimum thickness of 2 inches on the flat, increased to 3 inches on the sloping sides, extending down to the bottom of the main belt. It is of a uniform thickness of 2½ inches at the ends of the vessel.

The Armament and its Disposition.—The guns fitted are—Four 12-inch 45-caliber breech-loading guns, mounted in pairs in barbettes, two forward and two aft, with 10-inch armor on the walls. Four 10-inch 45-caliber breech-loading guns, mounted singly at each corner of the citadel in barbettes, with armor 6 inches thick. Twelve 6-inch 45-caliber breech-loading guns on pedestal mountings—ten on the main deck and two on the upper deck. Twelve 12½-pounder quick-firing guns, for repelling torpedo attack. Three 3-pounder quick-firing guns, with naval and boat mountings. Six Maxim rifle-caliber guns with landing and boat mountings. Five submerged 18-inch torpedo tubes.

The following table gives the ballistics and principal particulars of the larger guns:

Caliber, inches	12	10	6
Diameter of bore, inches	12	10	6
Length of bore, inches	540	450	270
Total length of gun, inches	556.5	464.6	280.07
Maximum pressure in chamber, tons	17	17	17
Weight of projectile, pounds	850	500	100
Total weight of gun, including the breech mechanism	*t. c. q.* 57 9 2	*t. c. q.* 35 3 2	*t. c. q.* 7 6 2
Muzzle velocity, feet per second	2,860	2,820	2,900
energy, foot-tons	48,210	27,570	5,830
Energy at four miles range, foot-tons	18,950	9,190	857

The Propelling Machinery.—The propelling machinery of the *Katori,* consists of two sets of four-cylinder triple-expansion engines balanced on the Yarrow-Schlick-Tweedy system, each set having one high, one intermediate and two low-pressure cylinders. The diameters of the cylinders are respectively 35½ inches, 56 inches and 63 inches, for each of the low-pressure cylinders, with a stroke of 48 inches. The steam pressure at the boilers is 230 pounds per square inch, and at the engines 200 pounds per square inch. Stephenson link motion is adopted for working the valves, which are of the piston type on the high-pressure and intermediate cylinders, and of the double-ported flat design on the low-pressure cylinders. The engines are designed to turn the propellers inwards when going ahead, so that the starting platform is in the center of the ship; wrought-steel columns form the supports of the cylinders, which are independent castings, and the back supports are of the ordinary cast-iron A framing, with ample slipper-guide surface. The condensers, which are four in number, are placed in the wings of the ship. The total cooling surface is 17,000 square feet. Each crank shaft is in two interchangeable pieces, and the propeller shaft is 18 inches in diameter, with a 10-inch hole, while the propellers have four blades, the diameter being 17 feet 3 inches. The blades and the boss are of Stone's bronze.

The boilers are of the latest Niclausse type, twenty in number, disposed in three separate boiler rooms, five with sixteen sections, and fifteen with fifteen sections, each section consist-

ing of twenty-four tubes. The total heating surface is 44,000 square feet, and the total grate area 1,334 square feet. There are two funnels, the forward one being 12 feet 9 inches in diameter over the casings, and the after one 12 feet 9 inches by 8 feet 1 inch over the casings, the height from the fire grate being 90 feet.

There is the usual complete system of pumps, with the addition of the independent air pumps, which are now a recognized improvement over pumps driven from the main engines.

Lighting, Heating and Ventilation.—The vessel (which will have a total complement of 980 officers and men) is lighted throughout by electricity and ventilated by electrically-driven fans. The thermo-tank system of ventilation has been adopted for the cabins and crew spaces and for ventilating, and also for cooling the magazines and shell rooms. A complete system of steam heating is fitted for the officers' quarters and the crew spaces. The steering engines and gear, electric generating plant, refrigerating, air-compressing and other auxiliary machinery, are all of the most improved types, and the vessel generally is fitted up in accordance with the most modern practice for ships of this class.

The Steam Trials.—The specification provided that the vessel should run at her fully-loaded draught for twenty-four hours at four-fifths of the designed power, in order to determine the coal consumption when steaming at what is regarded as the continuous speed in action. This trial took place on Wednesday and Thursday, April 25 and 26. The vessel, steaming right down the Irish Sea, experienced very rough weather; but proved very steady in a sea-way. On this trial, when the speed was 17.8 knots, the coal consumption worked out at 1.6 pounds per indicated horsepower per hour, which is proof alike of the ability of the Japanese stokers, after their long war experience, and of the general economy of the machinery. On Saturday, April 28, the vessel proceeded on her second trial, which, as prescribed, was to include four runs over a deep-sea course not less than 10 nautical miles long. The estuary of the Clyde offers splendid opportunities for such

tests, and two points were decided upon by the Japanese staff on board. Four runs between these points gave a mean of means speed of 20.22 knots, with the engines developing their full power, and making about 130 revolutions per minute. They continued at this rate of revolution in order to make a complete eight-hour's run. The trials were carried out, and all the data taken, by a Japanese staff, under the direction of the Special Commission in this country, which includes Captain Tanaka and Captain Fujii. The *Katori* was in command of Captain Sakamoto, who will take over the ship on behalf of his Government. The short period elapsing between the trials and the formal commissioning of the vessel, as well as the fact that she remains at her anchorage in the Firth of Clyde off Greenock during the process of completion, is evidence of the small amount of work remaining to be done, as a consequence of the inspection of the ship after the severe tests through which she has passed.—"Engineering."

The Gun Power and Speed of Battleships.—Japan's new battleship, the *Katori,* has, on her trials, marked a record in her combination of fighting power and speed. Built and completed in every respect for war by Messrs. Vickers Sons and Maxim, Limited, she represents what may be regarded as the highest conception, prior to the late war, of what a battleship should be; and the anticipations of the design have been more than fulfilled during the searching tests as to guns and speed made by the Japanese naval authorities in this country, terminating with most successful ordnance trials. The vessel is a compromise between the two present-day schools on the question of ordnance. Having four 12-inch and four 10-inch guns, all of 45 calibers in length, she has eight primary guns able to penetrate, at four miles range, the maximum thickness of armor on the broadside of any existing ships, for on trial the 12-inch pieces developed an energy at the muzzle of 48,000 foot-tons, and the 10-inch breech-loaders, of 27,570 foot-tons. There are four of these guns available for both bow and stern fire, and six for broadside attack. The feature is the adoption of the 10-inch guns in barbettes on the corners of the citadel,

and interest was centered in the tests as to rapidity of their fire. Improvements by the Vickers Company in breech mechanism, in the mountings, and in the system of supplying ammunition to the guns, proved most effective in both calibers of guns, and it was found that five shots were fired from the 10-inch weapons in 2 minutes 8.4 seconds. This result places the gun in a most advantageous position, as the rate of fire equals many pieces of lighter attacking power. These heavy pieces are unequal in number to those of some of the much larger ships now projected, for the *Katori* is under 16,000 tons displacement; but she possesses, in addition, twelve 6-inch guns, which are still favored because they facilitate concentration of fire owing to their quick-acting mechanism, and their fusilade of shell may paralyze the personnel on an enemy's ship, and thus indirectly win the day. The 6-inch guns in the *Katori* fired eight rounds in 52.4 seconds. The muzzle energy proved to be about 6,500 foot-tons. Thus the *Katori,* with a great number of small guns, is unequalled in power by any ship now in commission. Her speed is also unique, for on her eight-hours' full-power trial she averaged 20.22 knots, as determined by four runs over a long deep-sea course on the Firth of Clyde. On the twenty-four hours' coal-consumption test, at three-fourths of the full power, the mean was 1.6 pounds per indicated horsepower per hour. As she is most effectually armored, the *Katori* is thus a worthy addition to the victorious Navy of the Mikado, and appropriately comes from the works where Admiral Togo's flagship, the *Mikasa,* was constructed.—"Engineering."

PERU.

New British-Built Peruvian Cruiser.—A new twin-screw protected cruiser for the Navy of Peru, named *Almirante Grau,* has just been launched from the British shipyard of Vickers Sons & Maxim. This vessel has many features which will commend her to such naval powers as Peru, where the expenditure on the Navy is necessarily limited, as the vessel combines with the exceptionally high speed of 24 knots, a

considerable armament, including two weapons, each firing a 100-pound shell at a velocity effective against unarmored craft at three miles range. While thus able to engage such craft, she will, with the assistance of her high speed, greatly harass armored slower ships and will at the same time create those diversions which, when practised even by relatively weak fleets, materially influence the strategic plans of a superior naval force.

The *Almirante Grau* has a length of 370 feet, a breadth of 40 feet 6 inches, and a draught of 14 feet 3 inches, which will enable her to enter most of the harbors in South America. On this draught the displacement will be 3,200 tons.

The vessel, notwithstanding the limitations in size and cost, is effectively protected, as there extends a deck of 1½ inches thickness for the full length of the vessel, so as to cover in not only magazines and steering gear, but also the boiler and machinery compartments. Extensive coal bunkers are arranged along the sides in the wake of the machinery space, so as to minimize the penetration by shot and shell. The conning tower forward, from within which the ship will be controlled and fought, is constructed of nickel-steel hardened armor 3 inches thick.

As regards the armament, there is a 6-inch gun with an armored shield on the forecastle deck forward and one on the quarter deck aft. These guns, while primarily intended for bow and stern fire respectively, have a wide arc of training on the beam, the complete range being 270 degrees. The service of ammunition and projectiles to these guns has had careful consideration, in order to give them the maximum rapidity. Electric-motor ammunition hoists of the dredger type have been fitted, and these are protected by armor tubes. The secondary armament consists of eight 14-pounder quick-firing guns, and eight 1¼-pounder guns, distributed principally on the broadside, on the upper deck and bridge. There are two submerged tubes for firing 1-inch torpedoes.

The *Almirante Grau* will have a total complement of 300 officers and men, and the living quarters are more comfortable

than usual in such small craft. The fitting of the quarter deck aft has very considerably improved the officers' quarters, as compared with the preceding vessels of this type.

A complete system of natural and artificial ventilation has been adopted, and the ship is heated throughout by steam radiators. An electric installation supplies light for the entire ship. The admiral's and principal officers' cabins, as well as the wardroom, are not only spacious, but specially comfortable. A stern walk is provided, as in larger ships. The remaining officers are accommodated aft on the protective deck. The crew are berthed under the forecastle and upper deck forward.

Although the vessel is comparatively small, the conveniences are in excess of those fitted in much larger ships of a year or two ago for the British service. This applies not only to lavatories, but to baths, etc., which are included both for the officers and the crew, those for the latter being placed amidships on the protective deck. The vessel is also provided with refrigerating plant and ice-making machinery, with cold-storage chambers adjoining, and the bakery is equipped with mechanical dough kneaders, etc. Again, the vessel is fitted with hammock berthing instead of rails, and the stowage of the hammocks within these adds to the protection against machine-gun fire, as was almost universally the case in vessels for the British Navy a few years ago.

The machinery consists of two sets of four-cylinder triple-expansion engines, each set having four cranks balanced on the Yarrow-Schlick-Tweedy system. They are designed to give a collective indicated horsepower of 14,000, with a steam pressure of 250 pounds per square inch at the engines, and 280 pounds at the boilers. This will give a speed of 24 knots, and the coal endurance at cruising speed will be about 4,500 sea miles.

The cylinders of the main engines are carried on cast-iron A columns at the back, and polished-steel columns at the front, the whole being supported on a cast-steel bed plate. The valve gear, of the Stephenson type, with double eccentrics and link motion, actuates a piston valve in the case of the high-pressure

cylinder, two piston valves for each intermediate-pressure cylinder, and one double-ported flat slide valve for each low-pressure cylinder. Each low-pressure valve is provided with a relief arrangement and assistant cylinder. A two-cylinder reversing engine of the all-round type is fittted to each set. There is also for each set a single-cylinder turning engine. Each engine has one air pump worked by levers from the crosshead of the intermediate-pressure piston rod.

The crank and propeller shafting is of steel made by Messrs. Vickers Sons & Maxim, Limited, Sheffield, and is hollow. The propellers are three bladed, with detachable blades, and these, with the boss, are of bronze.

The main condensers (one for each engine) are of gun metal. Circulating water is supplied by one large centrifugal pump placed in each engine room.

The ten boilers, of the small-tube type, are arranged in three separate watertight compartments, and at full power will work under the closed-stokehold system by forced draft. The fans for supplying the air are driven by steam engines of the enclosed type arranged for forced lubrication. Feed pumps of the direct-acting type are fitted in the boiler rooms, and provision is made for dealing with the ashes by means of hydraulic ejectors and steam-driven hoists.

The vessel has a complete equipment of auxiliary machinery. The electric light, evaporating and distilling plant, and steam-steering engine are placed in a separate room aft of the main engine room.

The vessel is rigged with two masts, both of which have signal yards and gaffs. There is fitted an installation of wireless telegraphy. The *Almirante Grau* carries two powerful searchlights, and nine small boats will be taken, including a 34-foot steam launch. A sister vessel for Peru is in an advanced state of construction at Vickers' works.—"The Nautical Gazette."

MERCHANT SHIPS.

The Names of the High-Speed Cunard Liners.—The Cunard Company have now fixed upon names for their two 25-knot, quadruple-screw, turbine-driven liners, being built respectively by Messrs. John Brown & Co., Limited, of Clydebank, and Messrs. Swan, Hunter & Wigham-Richardson, Limited, of Wallsend-on-Tyne. The vessel by the first-named firm is to be called the *Lusitania,* which was the name of one of the old Roman provinces of the Spanish Peninsula, and until quite recently was regarded as practically identical with the present kingdom of Portugal; now, however, it has been found that the Roman province of the name lay wholly on the south side of the River Tagus. The Tyne-built vessel is to be called the *Mauritania,* which in old times was the name given to the northwestern part of Africa, opposite to Gibraltar, and corresponding now to Morocco and the western portion of Algiers.

The China Navigation Company's Steamer "Huichow."—The China Navigation Company, formed in 1873 by Messrs. John Swire & Sons, of London, have had a very considerable influence in the development of industry, and particularly of British commerce, in the Far East, in view of the extensive character of their transport work. They conduct traffic from China as far south as Australia, as far west as the Straits, and as far north as the Amur River, and have steamers trading up the Yangtze-Kiang a thousand miles from the sea, to Ichang, where the rapids prevent navigation farther into the interior. They have had built by the Scotts' Shipbuilding and Engineering Company, Greenock, sixty-four steamers, of an aggregate tonnage of 115,600 tons, and with engines of 14,967 nominal horsepower. These vessels have each marked steady progress; as early as 1878, owing to the strong advocacy of the late Mr. John Scott, C. B., the twin-screw system

of propulsion was adopted, and has since then been applied in many of the company's steamers.

The order for six vessels was placed in February, 1905; and as the first vessel was delivered in October, and the sixth in December, it will be recognized that a very smart piece of constructional work was done. The dimensions of the vessels are:

Length, 267 feet; breadth, 40 feet; depth, 22 feet 6 inches; gross tonnage (about), 2,000 tons; mean draught on trial, 9 feet 7 inches; displacement at this draught, 2,000 tons; revolutions on trial, 92; indicated horsepower, 1,730; steam pressure, 190 pounds; vacuum, 28 inches; speed, 12.91 knots.

Two decks are laid, the 'tween decks being lighted and ventilated with a view to carrying steerage passengers. The spar deck and the bridge deck are teak sheathed. Accommodation is provided for a number of first-class passengers. The staterooms are fitted with iron folding beds, electric fans, bells and light. In a house on the spar deck aft berths are provided for a large number of Chinese passengers, while the officers' and crew's accommodation is unusually extensive and commodious for a vessel of this tonnage. The arrangements for working the cargo are very complete, and are for two different systems of loading. There are six ordinary derricks and one for 25-ton lifts, with the necessary steam winches for working them.

In order to facilitate loading or discharging cargo from or to barges, large cargo posts are fitted on the ship's sides in the way of each hatch, and the holds have stages fitted for handling cargo down from man to man. For trimming the ship, and for use on light voyages, the fore peak and two small holds, or deep tanks, are fitted for carrying water ballast. The arrangements for working the ship include a steam windlass arranged for warping, a warping capstan, steam-steering gear, and hand-steering gear. Awnings are fitted fore and aft the ship, and other special arrangements are made for trading in tropical waters.

The engines are of the three-cylinder, inverted, triple-ex-

pansion type, and represent the practice of the constructors for intermediate steamers. The diameters of the cylinders are: High-pressure, 20 inches; intermediate-pressure, 33½ inches; low-pressure, 56 inches; the stroke being 39 inches. The cylinders are carried on cast-iron columns. The high-pressure cylinder only is jacketed, and is fitted with a piston valve; the intermediate-pressure and low-pressure cylinders have double-ported slide-valves. The valves are operated by the usual link motion. The valve spindles are of steel; the intermediate-pressure and low-pressure spindles are fitted with extension spindles, and have also, a special locking arrangement for locking the valves in position; the lower ends of the spindles are guided by a bracket with brass liner. The eccentric rods, quadrants and quadrant blocks are of steel, and are finished up bright throughout. The eccentric sheaves and straps are of cast iron.

All the pistons are of the conical type, and are of cast steel; the high-pressure piston is fitted with a solid cast-iron packing ring; the intermediate-pressure and low-pressure pistons have each two packing rings fitting into solid cast-iron case rings. The piston rods are of Siemens-Martin steel, and are bolted to steel crossheads, to which also are bolted the ahead and the astern guide slippers. These slippers are of cast iron, with white-metal guide faces. Water is circulated behind the ahead guide faces. The connecting rods are of wrought iron; the bottom end bushes are of cast steel, and have white-metal liners. The crank shaft is of the built-up type, and is made in three pieces, which are interchangeable; the material is Siemens-Martin steel, and the diameter of the shaft is 10¾ inches. The thrust block is of the usual type, with six shoes of the horseshoe pattern; these are adjusted by long steel adjusting screws and provision is made for oil service on each shoe. The propeller is of cast iron, with four blades; the diameter is 14 feet 3 inches and the pitch 14 feet.

The condenser, which has a cooling surface of 1,813 square feet, is cast along with the three columns at the back of the engine, and the exhaust from the low-pressure cylinder passes

to the condenser down the interior of the column of the intermediate cylinder. Water is circulated through the condenser by an independent centrifugal pump. The air pump (Edwards type), feed pumps, bilge pumps and a sanitary pump are connected to a common crosshead and worked by levers and links from the low-pressure engine. A combined feed heater and auxiliary condenser is fitted, into which the exhaust from the various auxiliaries pass.

Steam at a working pressure of 190 pounds is supplied by a single-ended cylindrical boiler 16 feet 7 inches in diameter and 12 feet 6 inches long. There are four furnaces, the mean internal diameter being 3 feet 9 inches. The total heating surface is 3,838 square feet, and the grate surface 90 square feet. A donkey boiler (Blake's patent) is also fitted for general purposes; this boiler works at a pressure of 90 pounds per square inch.

First Turbine Ship Built in America.—There was launched on Saturday, April 21, at Roach's Shipyard, Chester, Pa., the hull of the steamship *Governor Cobb,* now being built for the Eastern Steamship Co., of Boston, Mass., for their route between Boston and St. John, New Brunswick. The *Governor Cobb* is the first turbine-propelled steamship built in the United States. Her general dimensions are as follows: Length on water line, 290 feet; length over all, 300 feet; beam molded, water line, 51 feet; beam over guards, 55 feet; depth, 20 feet 6 inches; and draught, loaded, 14 feet.

The *Governor Cobb,* in general outside appearance, will be very similar to the steamships *Governor Dingley* and *Calvin Austin,* of the Eastern Steamship Co.'s fleet. She is a double-bottom steel ship, having main, saloon, gallery and dome decks. There will be about 175 staterooms and about 200 berths. The dining room will be placed on the main deck aft of quarter deck. The ship will be complete with all modern appliances, such as steam-steering gear, electric lights, electric call bells, etc., and special care will be taken in her construction and equipment to give full protection to her passengers.

The motive power will consist of Parsons marine steam

turbines, built by the W. & A. Fletcher Co., of Hoboken, N. J. There will be three propeller wheels, the center one being the high-pressure and the two outside ones being the low-pressure and astern turbines. The maximum working steam pressure will be 150 pounds. There will be six Scotch boilers, fitted for forced draft. The engines and boilers will be enclosed in steel up through and including the dome deck, so as to reduce the chance of fire from same as much as possible.

After the hull is launched she will be brought to the works of the W. & A. Fletcher Co. at Hoboken, N. J., who are the contractors for the ship complete, and will be completed ready for operation at their works. The Fletcher Co. sublet the building of hull to the Delaware River Iron Shipbuilding and Engine Works, Roach's Yard, Pa.

OBITUARY.

JOHN CHRISTIAN KAFER.

TRIBUTE OF AFFECTION BY WALTER M. MCFARLAND.

The recent death of this talented engineer and splendid man has removed one who played a very important part in the education of many naval officers, especially of those who passed through the separate course for engineers at the Naval Academy. Indeed all the cadet engineers, except those in the class of 1878, were at one time or another under his instruction.

Mr. Kafer was unusually fitted for the work of instruction, not only by his great natural ability and splendid equipment as an engineer, but above all by a generous sympathy with young men which lasted all through his life. Every man who had been fortunate enough to be one of his pupils counted him a personal friend, and many of us who were privileged to keep in close touch with him felt that we were specially honored when he permitted us to call him Uncle Jack.

Mr. Kafer was born December 27, 1842, so that, at his death on March 30, 1906, he was not yet sixty-four. When not yet quite 21, he entered the Navy in 1863, and was constantly "at the front" until the close of the Civil War, taking part in the James River campaign and in the first attack on Fort Fisher. In the natural routine of service he also made other cruises, notably one on the *Tennessee* (1875-8), where he was a close associate and friend of Admiral (then First Ass't Engineer) Melville; but his greatest service to the Navy was in his duty at Annapolis and at the Bureau of Steam Engineering. He was one of the first instructors in the Department of Steam Engineering at the Naval Academy when the separate course for engineers was formed, and altogether was an instructor for about ten years, broken by the cruise on the *Tennessee*. It is possible that the young engineers of today do

not realize fully how much credit is deserved by these older men who were the pioneers in technical education. At the present time the equipment of the technical schools and the literature of instruction are so complete that the personality of the instructor may almost be considered to occupy a secondary place. It was quite different with the men who were instructors in the sixties of the last century. There were practically no text books, and their courses of instruction had to be made up largely from their own experience and from their digest of technical literature. That they did splendid work is shown by the men who were turned out under such a course. Besides Mr. Kafer, the early corps of instructors in steam engineering at Annapolis included such men as Dr. Robert H. Thurston, Professor David M. Greene, John D. Van Buren, Chas. H. Manning, Wm. L. Nicoll, David Jones, Robert Crawford, George W. Roche, Frederick Shober, Thos. W. Rae and John Pemberton. Of those who were peculiarly identified by long service with the course, only Mr. Manning and Mr. Crawford remain. In connection with his work as a teacher, it should be mentioned that Mr. Kafer was invited in 1885 to become the Dean of Sibley College at Cornell University, but his health would not permit him to accept.

Mr. Kafer was principal assistant to Engineer-in-Chief Loring and also for a short time to Engineer-in-Chief Melville, to both of whom he was not only an able assistant but the most loyal of friends. This service covered the last of the old and the beginning of the new Navy, and in this work Mr. Kafer had a very prominent part. His health had not been good for many years, and indeed during a considerable part of his cruise on the *Tennessee* he stood watch on crutches on account of varicose veins in his legs, which made it almost impossible for him to keep on his feet for long periods. In March, 1888, he was retired on account of his physical condition, and the Navy thereby lost the active service of one of the finest men who ever held a commission, although it did not lose his heartfelt interest and active sympathy in every movement looking for an increase of efficiency.

After leaving the service he was a consulting engineer for a

time and then became General Manager, and later Vice President, of the Morgan Iron Works in New York, which position he held for a number of years until his desire for improvement of the plant, which was prevented by divided ownership of the stock, led to his severing his connection. For a time he was Vice President of the Quintard Iron Works, but his health was such that he could not tie himself down to any position which required uninterrupted service and he, therefore, relinquished this also. Very recently he formed a partnership with Messrs. Mattice and Warren, former pupils of his at Annapolis, and they had opened offices in New York as consulting engineers.

Mr. Kafer took a very active part in the technical societies connected with engineering. He had been a member of the Board of Managers and Vice President of the American Society of Mechanical Engineers and was a member of council of the Society of Naval Architects and Marine Engineers from its beginning. He attended the meetings whenever practicable and contributed materially to the value of the proceedings by taking part in the discussion of papers. At the Engineering Congress connected with the Chicago Exposition in 1903 he was one of the most active members of the Division of Marine Engineering and Naval Architecture of which his old Chief, Admiral Melville, was the Chairman. He was also the first American member of the Institution of Naval Architects of Great Britain, having joined very soon after the foundation of that society.

If he had done nothing more than the work which has been briefly summarized above, he could have looked upon his life as well spent and very useful, but besides this he was for many years the most active spirit in the Engineers' Club of New York, which has become, possibly, the greatest factor of what might be called the social side of engineering. He was one of the earliest members of the Club when it occupied quarters on Twenty-ninth Street, and it was due almost entirely to his courage and persistence that the Club moved to its present fine quarters on Fifth Avenue. He was Chairman of the House

Committee when the removal occurred, and was chiefly responsible for the adaptation of the building to Club purposes. He was President of the Club from 1901 to 1904, and had been for many years one of the Board of Governors. For some years he had realized, more fully than perhaps anyone else, that the growth of the Club would compel it to seek new quarters and preferably in a building of its own. At his own risk he secured options for the splendid site on West Fortieth Street facing Bryant Park where the beautiful new home of the Club, the gift of Mr. Andrew Carnegie, is now under erection. The building of this new club house was very near his heart, and he had been, from the start, the Treasurer of the Building Committee, and was giving the work of supervision a very large part of his time. He was also Treasurer of the Building Committee of the United Engineering Building on Thirty-ninth Street, another of Mr. Carnegie's gifts. In fact he was Mr. Carnegie's representative in the financial side of these undertakings. He was not spared to see their completion, but those who know the part which he played will always feel that the new Engineers' Club is a monument to him.

With all his amiability and sympathy Mr. Kafer was a man of very strong character and one whose courage in any cause which he championed never faltered. In the old days of the unfortunate line and staff fight in the Navy, he was one of the foremost champions of the cause of the engineers, but it is very pleasant to note that so reasonable and conscientious was he in all his efforts that he enjoyed in the highest degree the respect and confidence of his brother line officers, many of whom were among his intimate friends.

Within the past year a number of his friends and admirers in the Engineers' Club purchased a fine portrait of him by one of the foremost artists in New York and presented it to the Club. It is pleasant to think that this evidence of the affectionate regard in which he was held came in time for him to enjoy it.

A touching tribute came, on the day of his death, in the form

of a telegram from that beloved old man and famous engineer, John Fritz, now in his 85th year, to the Club, saying: "I sincerely sympathize with you all in the loss of John C. Kafer. We have lost a dear friend, the club a valuable member, the country a good citizen and a useful man."

As a personal friend he was ideal. His attractive personality, lovable disposition, broad sympathy and tactfulness drew men to him, so that his companionship was always a pleasure, and any gathering that included him was the more genial and successful for his presence. His kindness and encouragement to younger men meant a great deal to them, and those who knew him well will long feel a heartache at the loss of Uncle Jack. No words of praise can adequately express the love which his friends bore him, but it can be truly said that a most lovable, high-minded and public spirited man, as well as an accomplished engineer has passed away, leaving a record which all may cherish and which is without a stain.

LIEUTENANT JOHN M. HUDGINS, U. S. NAVY.

Lieutenant John M. Hudgins, U. S. Navy, who lost his life, April 13, 1906, by an explosion on board the *Kearsarge* during target practice at Guantanamo, Cuba, was born in Virginia, November 22, 1871, and appointed to the Naval Academy from that State, September 8, 1890. He graduated in 1894, and was assigned to engineering duties.

He was identified with the development of the wireless telegraph system now used in the Navy, having been attached to the Bureau of Equipment during the experiments and inauguration of the system.

The death of Lieutenant Hudgins is a distinct loss, not only to our Society, but to the Navy as a whole.

BOOKS RECEIVED.

A PRACTICAL MANUAL OF TIDES AND WAVES.—By W. H. WHEELER, M. Inst. C. E. New York and London: LONGMANS, GREEN AND COMPANY. Price $2.80.

The author has given a practical account of the action of the sun and moon in producing tides, without going into mathematical demonstrations.

The laws relating to the formation of tidal currents as distinguished from tidal waves are explained, and their effect on navigation dealt with. No attempt is made to deal with the subject of tides in a scientific way.

The book is clearly written and contains much information useful to the navigator and of interest to the student.

It is arranged in fifteen chapters, with appendices, as follows:

Chapter I. Introduction.—II. Development of Tidal Science.—III. The Sun, the Moon, and the Earth.—IV. The Making of the Tides.—V. Propagation of the Tidal Wave.—VI. Tidal Currents.—VII. Mean Level of the Sea and Range of the Tides.—VIII. Effect of the Wind and Atmospheric Pressure on the Tides.—IX. River Tides.—X. Wind Tides.—XI. Seismic and Cyclonic Storm Waves.—XII. Tidal Bores in Rivers.—XIII. Tide Tables and Tide Gauges.—XIV. Secondary Undulations and Storm Warnings.—XV. The Tides as a Source of Power.

There are eight appendices containing useful data and references.

ASSOCIATION NOTES.

At a meeting of the Council of the Society held on Saturday, March 17, 1906, at the Navy Department, the votes of the members for the prize essay were opened and counted.

The count showed that the majority of the members did not favor awarding a prize for 1905, and the Council therefore decided that no prize should be given.

The Council determined that, in future, the authors (who must be members) of original articles of special merit, contributed to the JOURNAL, should be paid for their work, the amount to be fixed by the Council in each case.